

# APPLIED HEAT TRANSMISSION

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BY

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# APPLIED HEAT TRANSMISSION

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## PREFACE

This book is intended for both practicing engineers and students. Its purpose is to put into readily usable form some of the more important data on heat transmission and to describe some of the common types of heat-transfer equipment and kinds of insulation used in industry. As indicated by the title, emphasis is placed on practical applications rather than on theory.

The first three chapters deal with the equations required for the solution of elementary problems involving heat transfer by conduction, radiation, and convection. It is hoped that the method of presentation used is one that will be particularly attractive to practicing engineers, who are likely to be more concerned with methods of solution than with mathematical derivations. Thus, in each section the equations are stated first, the methods of applying them are then illustrated by the solution of numerical problems, and finally the derivation of the equations is explained.

Attention is directed particularly to the charts and tables presented in Chaps. IV and V. By means of these charts and tables, convection coefficients and values of the pressure drop for a number of fluids under various conditions can be quickly and easily determined. Although the accuracy of some of the equations upon which these charts and tables are based is not yet definitely established, it is believed that in every case the values given are conservative and may be safely used for design purposes. The method of applying these charts and tables is illustrated by the solution of numerical problems at the end of each chapter.

The last two chapters are devoted to a description of some of the more common types of heat-transfer equipment and kinds of insulation used in industry. Equations for calculating the thickness of insulation required for several special conditions of industrial importance are given at the end of the last chapter.

The application of these equations is also illustrated by the solution of several numerical problems

Many of the equations upon which the charts and tables given in Chaps IV and V are based are taken from "Heat Transmission," by Mr William H McAdams, and indebtedness to this author is gladly acknowledged Most of the calculations required for the preparation of these charts and tables were made by Messrs A Peter Alyea and Adolph O Lee, and most of the illustrations throughout the book were prepared by Mr John M Hobbs It is with particular pleasure that the valuable assistance of these men is acknowledged The courtesy of numerous manufacturers who supplied photographs and drawings is also acknowledged Finally, the assistance and encouragement offered by Prof M P Cleghorn of Iowa State College are gratefully acknowledged

H J STOEYER

AMES IOWA  
January 1911

# CONTENTS

	PAGE
PREFACE. . . . .	V
INTRODUCTION . . . . .	1
CHAPTER I	
CONDUCTION . . . . .	3
CHAPTER II	
RADIATION. . . . .	23
CHAPTER III	
CONVECTION . . . . .	37
CHAPTER IV	
CONVECTION COEFFICIENTS. . . . .	54
CHAPTER V	
CALCULATION OF PRESSURE DROP. . . . .	113
CHAPTER VI	
TYPES OF INDUSTRIAL HEAT-TRANSFER EQUIPMENT . . . .	136
CHAPTER VII	
INSULATION. . . . .	162
APPENDIX . . . . .	209
INDEX. . . . .	217

# APPLIED HEAT TRANSMISSION

## INTRODUCTION

Heat may be transferred by conduction, by convection, and by radiation. These three methods may be defined as follows:

**Conduction** is the transfer of heat from one particle of matter to another, the particles remaining in fixed positions relative to each other.

**Convection** is the transfer of heat from one part of a fluid to another by the mixing of the warmer particles of the fluid with the cooler. For example, consider a pan of water placed on a hot stove. Heat is transferred through the pan and to the particles of water in contact with the pan by conduction. These particles, becoming more buoyant as they are heated, rise and mix with the main body of the water, thus transferring heat by convection. Such motion of the fluid, caused entirely by differences in density within the fluid, is called *natural convection*. If the motion of the fluid is produced by some mechanical means, such as a stirrer, pump, or fan, it is called *forced convection*.

**Radiation** is the transfer of heat from one body to another as the result of the bodies' emitting and absorbing a form of energy called *radiant energy*. All matter possesses the property of both emitting and absorbing such energy to a greater or lesser degree, the effect on the matter being the same as though heat were absorbed from or added to it.<sup>1</sup> Radiant energy travels through space at the velocity of light, light being radiant energy of those wave lengths to which the human optic nerve is sensitive. Unlike heat, radiant energy does not require the presence of any matter for its transmission. Thus, the earth receives radiant energy from the sun even though separated from it by an almost perfect vacuum.

<sup>1</sup> Except in the rare cases where photochemical reactions are involved

In practice, heat transfer by more than just one of the three methods is usually involved. For example, consider the heat transfer from an ordinary steam "radiator" to the air in a room. The flow of heat through the metal walls of the radiator and to the air in contact with the surface of the radiator takes place by conduction. As the air surrounding the radiator is heated, it rises and carries heat to the rest of the room by convection. At the same time the radiator emits radiant energy, most of which is transmitted through the air to the walls of the room. The walls absorb part of this radiant energy and convert it into heat and reflect the rest back to the radiator or to other parts of the wall. The total rate at which the radiator loses heat is equal to the rate at which it loses heat by convection plus the rate at which it loses heat by radiation, and this total rate is necessarily equal to the rate at which heat flows through the walls of the radiator by conduction.

Since all three methods of heat transfer are likely to be involved, it is necessary to be able to calculate the rate of heat transfer by each of the three methods in order to design heat-transfer equipment. It frequently happens, however, that even though all three methods may be involved, the rate of heat transfer by one of the methods is small and may be neglected. For example, the effect of radiation may usually be neglected in designing shell-and-tube type equipment.



# CHAPTER I

## CONDUCTION

✓ 1. **Thermal Conductivity.**—The calculation of the rate at which heat flows through any material by conduction involves a property of the material called the *thermal conductivity*. This property is defined as the *rate* at which heat flows through the material by conduction per unit of cross-sectional area taken normal to the direction of heat flow and per unit temperature gradient measured in the direction of heat flow. Consider, for example, a small element of the material having a cross-sectional

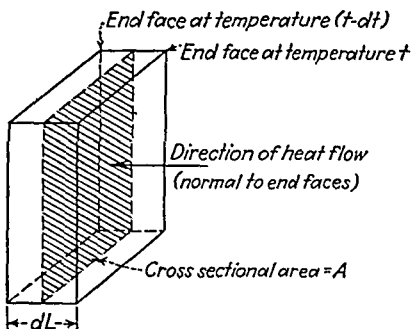


FIG. 1.—Unidirectional conduction of heat.

area  $A$  and thickness  $dL$ , as shown in Fig. 1. If the two end faces of the element are maintained at the uniform temperatures  $t$  and  $(t - dt)$ , and if the sides of the element are surrounded by other material at the same temperature so that no heat is lost laterally, heat will flow at some rate  $q$  from the face at the higher temperature to the face at the lower temperature. Since the end faces are both at uniform temperatures, the direction of heat flow will always be normal to these faces. By definition, the thermal conductivity  $k$  of the material is equal to the rate of heat flow  $q$  divided by  $A$  (the cross-sectional area normal to the direction of heat flow) and divided by  $-dt/dL$  (the temperature gradient in the direction of heat flow); i.e.,

$$k = \frac{q}{A(-dt/dL)} \quad (1)$$

The thermal conductivities of various materials differ widely and can be determined only experimentally. Tables of thermal conductivities of various materials are given in the Appendix.

The units of  $k$  depend upon the units chosen for each of the quantities in the right-hand member of Eq. (1). Thus, if the rate of heat flow  $q$  is measured in B t u per hour, if the cross sectional area  $A$  is measured in square feet, and if the temperature gradient  $dt/dL$  is measured in degrees Fahrenheit per foot, the net units of  $k$  are

$$\frac{\frac{\text{B t u}}{\text{hr}}}{\text{ft}^2 \times \frac{^\circ\text{F}}{\text{ft}}} = \frac{\text{B t u}}{(\text{ft})(\text{hr})(^\circ\text{F})}$$

These are the units that are used throughout the book. The thermal conductivities of many engineering materials, particularly industrial insulating materials, are frequently expressed in the units B t u / (hr) (sq ft) (°F / in). However these units are inconvenient in calculating the rate of heat transfer by conduction through bodies of any shape other than flat. Values of  $k$  expressed in such units can be converted to the units B t u / (ft) (hr) (°F) by dividing them by 12.

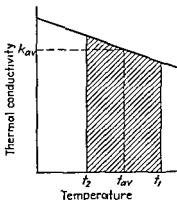


FIG. 2—Variation of thermal conductivity with temperature

**2 Effect of Temperature and Pressure on Thermal Conductivity**—The thermal conductivities of practically all materials

depend upon the temperature of the material. For some materials the thermal conductivity increases as the temperature of the material rises; for others it decreases. This variation is approximately linear over a considerable range of temperature for most materials. Thus, if the thermal conductivity is plotted against temperature as in Fig. 2, a straight line is obtained.

For practical purposes the thermal conductivities of all materials are unaffected by changes in pressure. At extremely

high pressures the thermal conductivities of liquids do increase, and at very low absolute pressures the thermal conductivities of gases decrease; but these pressures are outside the range usually encountered in engineering work.

**3. Steady and Unsteady Conduction of Heat.**—The conduction of heat through any body is said to be *unsteady* if the temperature at any point in the body varies with time, and it is said to be *steady* if the temperature at every point in the body remains constant. Hence, with unsteady conduction the temperature gradient  $dt/dL$  at each point in the body also varies with time; with steady conduction the temperature gradient at each point remains constant. For example, when steam is first admitted to a pipe covered with insulation, the temperature at the inner surface of the insulation rises very rapidly, but the temperature at any point near the outer surface is not immediately affected. The temperature at each point in the insulation continues to rise for some time, and during this period the flow of heat through the insulation is unsteady. Finally, however, the temperature at each point reaches some value at which it remains constant, and the flow of heat is then steady.

**4. Fundamental Equation for Conduction.**—Equation (1) can be written

$$q = -kA \frac{dt}{dL}. \quad (2)$$

This equation, known as *Fourier's law*, is the fundamental equation for unidirectional heat transfer by conduction. The equation applies to fluids as well as to solids, provided that no convection currents exist within the fluids. It is impossible to obtain this condition, however, except in very thin layers of the fluid, and consequently the rate of heat transfer through fluids must ordinarily be calculated by the equations for convection given in Chap. III.

Equation (2) must be integrated before it can be used to solve practical problems. If the conduction of heat is unsteady, this integration is usually quite difficult, because the temperature gradient  $dt/dL$  at each point in the body varies with time, and therefore the rate of heat flow  $q$  also varies with time. Fortunately, many practical problems involve only steady conduction. For steady conduction, the rate of heat flow  $q$  is constant because the temperature gradient  $dt/dL$  at each point in the body

remains constant. Consequently, for steady conduction the integration of Eq (2) requires only (1) that the area  $A$  of the isothermal surfaces existing within the body be expressed as a function of the distance  $L$  to these surfaces and (2) that the thermal conductivity  $k$  of the body be expressed as a function of the temperature  $t$ . Even this integration is difficult, however, unless the area  $A$  is some simple function of the distance  $L$ .

Integrated forms of Eq (2) for the steady conduction of heat through several types of bodies frequently encountered in practice are given in the next three sections. These equations all involve the surface temperatures of the bodies. In practice these temperatures are frequently not known, but instead only the temperatures of the media in contact with the surfaces of the body are known. For example, in the case of insulation on piping the temperature of the air surrounding the pipe is always known, but the temperature of the outside surface of the insulation itself is usually not known. Surface temperatures can be determined by making use of the fact that, for steady conduction the rate at which heat is transferred through the body by conduction is equal to the rate at which heat is transferred from the surfaces of the body to the surrounding media by radiation and convection. This method of calculation is explained in detail in Sec 61 in Chap VII. In the present chapter it is assumed that the surface temperatures of the bodies are known.

An approximate method of calculating temperature gradients and values of  $q$  for the unsteady conduction of heat through flat bodies is explained in Sec 10.

✓ **5 Conduction of Heat through Flat Bodies** —If the conduction of heat is steady, the rate of heat transfer through homogeneous flat bodies can be calculated by the equation

$$q = \frac{kA(t_1 - t_2)}{L}, \quad (3)$$

where  $q$  = the rate of heat transfer by conduction, B t u per hr

$k$  = the thermal conductivity of the material at the average of temperatures  $t_1$  and  $t_2$  B t u / (ft) (hr) (°F)

$A$  = the cross-sectional area of the body, taken normal to the direction of heat flow, sq ft

$t_1$  and  $t_2$  = the temperatures at the two faces of the body, °F

$L$  = the thickness of the body, ft

**Illustrative Problem 1.**—Calculate the rate of heat transfer by conduction through each square foot of a wall 8 in. thick made of calcined, diatomaceous silica insulating bricks if the temperatures at the two faces of the wall are 1200 and 200°F.

**Solution.**—The average of the temperatures at the two faces of the wall is 700°F., and the thermal conductivity of calcined, diatomaceous silica bricks at this temperature is 0.15 B.t.u./(ft.)(hr.)(°F.) Hence

$$q = \frac{0.15 \times 1 \times (1200 - 200)}{\frac{8}{12}} \\ = 225 \text{ B.t.u. per hr.}$$

Equation (3) can be obtained from Eq. (2) as follows: Since steady conduction is assumed,  $q$  is constant; and since the body is flat,  $A$  is constant. Hence, Eq. (2) can be written

$$\frac{q}{A} \int_0^L dL = - \int_{t_1}^{t_2} k dt.$$

The right-hand member is represented by the shaded area shown in Fig. 2. If  $k$  is assumed to vary linearly with temperature, this area is equal to  $k_{av}(t_1 - t_2)$ , where  $k_{av}$  is the value of  $k$  at the arithmetic average of  $t_1$  and  $t_2$ . Therefore, integrating,

$$\frac{q}{A} L = k_{av}(t_1 - t_2),$$

which is Eq. (3).

**6. Conduction of Heat through Cylindrical Bodies.**—If the conduction of heat is steady, the rate of heat transfer through homogeneous bodies whose surfaces are *concentric cylinders* can be calculated by the equation

$$q = \frac{k2\pi l(t_1 - t_2)}{2.3 \log_{10} (r_2/r_1)}, \quad (4)$$

where  $q$  = the rate of heat transfer by conduction from the inner to the outer surface, B.t.u. per hr.

$k$  = the thermal conductivity of the material at the average of temperatures  $t_1$  and  $t_2$ , B.t.u./(ft.)(hr.)(°F.).

$\pi$  = 3.14.

$l$  = the length of the cylinder, ft.

$t_1$  and  $t_2$  = the temperatures at the inside and outside faces respectively, °F.

$r_1$  and  $r_2$  = the radii (or the diameters) of the inside and outside faces respectively, ft. or in.

If  $t_2$  is greater than  $t_1$ , a negative value is obtained for  $q$ , which indicates that the direction of heat flow is from the outer to the inner surface

It is interesting to note that since only the ratio of the radii appears in Eq (4), the rate of heat transfer by conduction per foot of length is independent of the absolute size of the cylinder. Thus, if the surface temperatures of the insulation are the same, the rate of heat loss per foot of length from a 1-in diameter tube covered with  $\frac{1}{2}$  in of insulation is the same as the rate of heat loss from a 6 in diameter tube covered with 3 in of the same insulation

It is frequently possible to use Lq (3) in place of Eq (4) without introducing any serious error. Thus if the ratio of  $r_2/r_1$  does not exceed 2, substituting an arithmetic average of the inside and outside areas for  $A$  in Eq (3) will give a result within 4 per cent of that given by Eq (4)

**Illustrative Problem 2**—Calculate the rate of heat loss per foot of length from a 3-in pipe covered with 1 in of 85% Magnesia if the temperature of the inside surface of the insulation is  $500^\circ\text{F}$  and the temperature of the outside surface is  $100^\circ\text{F}$

**Exact Solution**—The average of the temperatures of the inside and outside surfaces of the insulation is  $300^\circ\text{F}$  and the thermal conductivity of 85% Magnesia at this temperature is  $0.043 \text{ Btu}/(\text{ft})(\text{hr})(^\circ\text{F})$ . The actual outside diameter of a 3-in pipe is 3.500 in and therefore the outside diameter of the insulation is 5.500 in. Hence by Eq (4)

$$q = \frac{0.043 \times 2 \times 3.14 \times 1 \times (500 - 100)}{2.3 \log_e (5.500/3.500)} \\ = 240 \text{ Btu per hr}$$

**Approximate Solution**—Since the ratio of  $r_2/r_1$  is less than 2 the rate of heat loss can be calculated approximately by Eq (3)

$$q = \frac{0.043 \times 3.14 \times (4.500/12) \times 1 \times (500 - 100)}{1/12} \\ = 243 \text{ Btu per hr}$$

Equation (4) can be obtained from Eq (2) as follows. The area  $A$  of any cylindrical element taken as shown in Fig 3 is equal to  $2\pi L l$ . Since steady conduction is assumed  $q$  is constant and Eq (2) can be written

$$\frac{q}{2\pi l} \int_{r_1}^{r_2} \frac{dL}{L} = - \int_{t_1}^{t_2} k dt$$

As in Sec. 5, if  $k$  varies linearly with the temperature, the right-hand member is equal to  $k_{av}(t_1 - t_2)$ . Hence, integrating,

$$\frac{q}{2\pi l} \log_e \left( \frac{r_2}{r_1} \right) = k_{av}(t_1 - t_2),$$

which is Eq. (4).

7. **Conduction of Heat through Spherical Bodies.**—If the conduction of heat is steady, the rate of heat transfer through

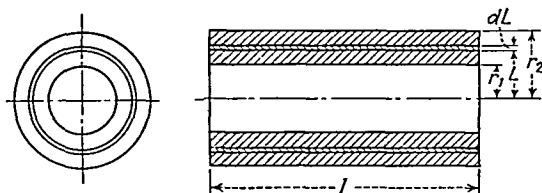


FIG. 3.—Cross section of a homogeneous, hollow cylindrical body.

homogeneous bodies whose surfaces are *concentric spheres* can be calculated by the equation

$$q = \frac{k4\pi r_1 r_2 (t_1 - t_2)}{r_2 - r_1}, \quad (5)$$

where  $q$  = the rate of heat transfer by conduction from the inner to the outer surface, B.t.u. per hr.

$k$  = the thermal conductivity of the material at the average of temperatures  $t_1$  and  $t_2$ , B.t.u./(ft.)(hr.)(°F.).

$\pi = 3.14$ .

$t_1$  and  $t_2$  = the temperatures at the inside and outside faces respectively, °F.

$r_1$  and  $r_2$  = the radii of the inside and outside faces respectively, ft.

As with Eq. (4), a negative value of  $q$  indicates that the direction of heat flow is from the outer to the inner surface.

It may be noted that, whereas the rate of steady conduction through either flat or cylindrical bodies is zero if the bodies are infinitely thick, the rate of steady conduction through spherical bodies never becomes less than a certain limiting value, regardless of how thick the walls may be. Thus, if  $r_2$  is infinite, Eq. (5) reduces to

$$q = k4\pi r_1 (t_1 - t_2) \quad (5a)$$

**Illustrative Problem 3**—Liquid oxygen is frequently stored in spherical containers because the ratio of outside surface to volume is a minimum for a sphere. (1) Calculate the rate at which heat leaks into such a container 5 ft in diameter insulated with a 1 ft thick layer of powdered diatomaceous silica packed to a density of approximately 10 lb per cu ft, if the inner surface of the insulation is at  $-290^{\circ}\text{F}$  and the outer surface is at  $50^{\circ}\text{F}$ . (2) What minimum rate of heat leak could be obtained by increasing the thickness of the insulation?

**Solution**—1 Since the thermal conductivities of all porous insulating materials decrease as the temperature of the insulation is lowered, the thermal conductivity of powdered diatomaceous silica packed to the given density and at an average temperature of  $-120^{\circ}\text{F}$  is less than the value  $0.026 \text{ Btu}/(\text{ft})(\text{hr})(^{\circ}\text{F})$  at  $70^{\circ}\text{F}$ . Assuming a value of  $0.022 \text{ Btu}/(\text{ft})(\text{hr})(^{\circ}\text{F})$ ,

$$q = \frac{0.022 \times 4 \times 3.14 \times 2.5 \times 3.5 \times (-290 - 50)}{1}$$

$$= -820 \text{ Btu per hr},$$

the negative sign indicating that the direction of heat flow is into the container.

2 If the insulation were made infinitely thick, the rate of heat leak would be, by Eq (5a),

$$q = 0.022 \times 4 \times 3.14 \times 2.5 \times (-290 - 50)$$

$$= -235 \text{ Btu per hr}$$

Equation (5) can be obtained from Eq (2) in a manner exactly analogous to that used in deriving Eq (4).

**8. Conduction of Heat through Bodies of Any Shape**—The integration of Eq (2) for homogeneous bodies of any shape other than those discussed in the three preceding sections is quite difficult, and consequently it is usually impossible to calculate the exact rate of heat transfer by conduction through such bodies. A satisfactory approximation can frequently be obtained, however, by using an arithmetic average of the inside and outside areas for  $A$  in Eq (3). This procedure gives too high a value for the rate of heat transfer, although the error is small if the thickness of the body (and consequently the difference between the inside and outside areas) is relatively small. The importance of this error is frequently minimized by the presence of other inaccuracies. Thus, the thermal conductivity of the material may not be accurately known, the body may not be perfectly homogeneous, or the temperatures at the two faces may not be absolutely uniform.

**Illustrative Problem 4**—A cylindrical tank 15 in in diameter and 5 ft long is insulated with 3 in of felted rock wool. If the inside surface of the



insulation is at 640°F. and the outside surface is at 120°F., calculate the rate at which heat is lost from the tank.

*Approximate Solution* —The inside area of the insulation is 22 0 sq ft; the outside area is 35 0 sq ft; and the average of these two areas is 28 5 sq ft. The average of the inside and outside surface temperatures is 380°F.,



FIG 4 —Composite wall of a brick kiln in process of construction (Courtesy of the Johns-Manville Co)

and the thermal conductivity of rock wool at this temperature is 0 045 B t u /(ft )(hr )(°F) Hence, by Eq (3),

$$q = \frac{0.045 \times 28.5 \times (640 - 120)}{\frac{3}{12}} \\ = 2670 \text{ B t u per hr.}$$

For the special case of rectangular bodies having relatively thick walls, as in small furnaces, an average area to be used in Eq. (3) can be calculated by empirical equations given by Langmuir.<sup>1</sup>

<sup>1</sup> *Trans Am Electrochem Soc*, vol 24, p 53, 1913

### 9 Conduction of Heat through Several Bodies in Series —

Furnace walls, The insulation on high temperature piping, and many other types of equipment are frequently constructed of layers of several different kinds of materials through which heat flows in series by conduction. For example, Fig 4 shows a brick kiln wall consisting of an inner layer of firebrick, a core of insulating brick, and an outer layer of red brick. The firebrick is used to protect the insulating brick from mechanical abrasion

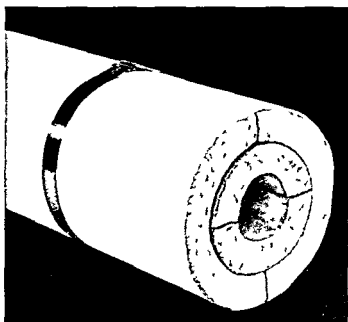


Fig 5—Combination high temperature pipe insulation (Courtesy of the Keesee & Mattison Co)

and from the high temperatures existing within the kiln. The red brick is cheaper than the insulating brick but has a higher thermal conductivity and cannot be used at the high temperatures existing at the center of the wall. Figure 5 shows a type of insulation suitable for covering high-temperature piping. It consists of an inner layer of high temperature insulation and an outer layer of 85% Magnesia. The 85% Magnesia has a lower thermal conductivity than the high temperature insulation but cannot be used at temperatures above 600°F. The rate of

heat transfer by conduction through such composite bodies can be calculated as follows:

If the conduction of heat is steady, the rate of heat transfer through two *flat bodies* in series, as shown in Fig. 6, can be calcu-

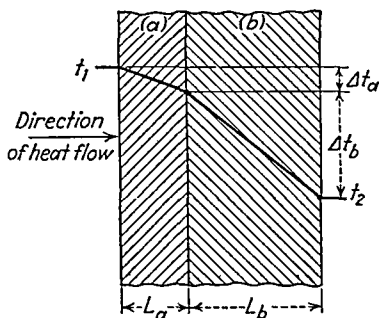


FIG. 6.—Temperature gradient through two flat bodies in series.

lated by the equation

$$q = \frac{A(t_1 - t_2)}{(L_a/k_a) + (L_b/k_b)}, \quad (6)$$

where  $q$  = the rate of heat transfer by conduction, B.t.u. per hr.  
 $k_a$  and  $k_b$  = the thermal conductivities of materials  $a$  and  $b$ ,  
 evaluated at the average temperature of each,  
 B.t.u./(ft.)(hr.)(°F.).

$A$  = the cross-sectional area of the bodies, taken normal  
 to the direction of heat flow, sq. ft.

$t_1$  and  $t_2$  = the temperatures at the outside faces of the com-  
 posite body, °F.

$L_a$  and  $L_b$  = the thicknesses of materials  $a$  and  $b$ , ft.

This equation may also be written

$$q = \frac{(t_1 - t_2)}{R_a + R_b}, \quad (6a)$$

where  $R_a$  and  $R_b$  are equal to  $L_a/k_aA$  and  $L_b/k_bA$ , respectively, and  
 are called the *resistances* of the two materials. These two equa-  
 tions can be extended to include any number of flat bodies in  
 series by adding additional  $L/k$  terms to the denominator of  
 Eq. (6) or resistances  $R$  to the denominator of Eq. (6a).✓

If the conduction of heat is steady, the rate of heat transfer  
 through two *cylindrical bodies* in series, as shown in Fig. 7, can  
 be calculated by the equation

$$q = \frac{2\pi l(t_1 - t_2)}{2.3 \left[ \frac{\log_{10}(r_a''/r_a')}{k_a} + \frac{\log_{10}(r_b''/r_b')}{k_b} \right]} \quad (7)$$

where  $q$  = the rate of heat transfer by conduction from the inner to the outer surface, B t u per hr

$k_a$  and  $k_b$  = the thermal conductivities of materials  $a$  and  $b$ , evaluated at the average temperature of each, B t u / (ft) (hr) ( $^{\circ}$ F).

$\pi = 3.14$

$l$  = the length of the cylinders, ft

$t_1$  and  $t_2$  = the temperatures at the inside and outside faces of the composite body,  $^{\circ}$ F

$r_a'$  and  $r_a''$  = the radii (or the diameters) of the inside and outside faces of material  $a$ , ft or in

$r_b'$  and  $r_b''$  = the radii (or the diameters) of the inside and outside faces of material  $b$ , ft or in

This equation can be extended to include any number of con-

centric cylindrical bodies in series by adding additional  $(1/k) \log_{10}(r''/r')$  terms to the denominator.

In applying Eqs (6) or (7), the thermal conductivity of each of the materials should be evaluated at the average temperature of the material, and consequently the temperatures between the various layers must be determined. These temperatures may also be required for design purposes if some of the materials are suitable for only a limited range of temperature.

They can be determined as follows: (1) A reasonable value for each temperature is first assumed. (2) Based on these assumed values, the thermal conductivity of each of the materials is determined. (3) The rate of heat transfer  $q$  through the composite body is calculated by Eq (6) or (7). (4) Using this value of  $q$ , the temperatures at each of the interfaces is calculated by Eq (3) or (4). These calculated temperatures will usually be sufficiently accurate for practical purposes, although the procedure may be

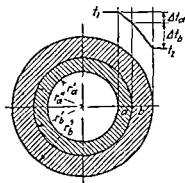


FIG. 7—Temperature gradient through two cylindrical bodies in series

repeated if greater accuracy is desired. The method of calculation is illustrated in the following problems:

**Illustrative Problem 5.**—A furnace wall consists of 9 in. of firebrick covered with 4 in. of insulating blocks made of diatomaceous silica, asbestos fiber, and a bonding material. Calculate the rate of heat transfer by conduction through each square foot of the wall if the temperature at the inside face is 2000°F. and the temperature at the outside face is 200°F.

*Solution.*—As a first approximation, assume that the temperature at the interface between the two materials is 1600°F. Based on this assumption, the average temperatures of the firebrick and of the insulation are 1800 and 900°F., respectively, and the thermal conductivities of the two materials at these temperatures are 0.71 and 0.061 B.t.u./(ft.)(hr.)(°F.). By Eq. (6), the rate of heat transfer through the composite wall is

$$q = \frac{1 \times (2000 - 200)}{\frac{9}{12 \times 0.71} + \frac{4}{12 \times 0.061}} \\ = 276 \text{ B.t.u. per hr.}$$

Since this is also the rate of heat transfer through each individual material, a more accurate value of the temperature  $t'$  at the interface can be calculated by Eq. (3). Thus, for the insulation,

$$276 = \frac{0.061 \times 1 \times (t' - 200)}{\frac{4}{12}}$$

whence

$$t' = 1710^\circ\text{F.}$$

The same result would be obtained by applying Eq. (3) to the firebrick. This temperature does not agree very closely with the temperature originally assumed, but an appreciably different value of  $q$  would not be obtained by repeating the calculations, since the thermal conductivities of the two materials vary only slightly with the temperature. Thus, a value of 280 B.t.u. per hr. is obtained if the calculations are repeated.

**Illustrative Problem 6.**—A 2-in. pipe is to be covered with two layers of insulation each 1 in. thick. An insulating material made of diatomaceous silica, asbestos, and a bonding material is to be used for the inner layer; and 85% Magnesia, which is suitable only for temperatures up to 600°F., is to be used for the outer layer. Will this covering be satisfactory if the temperature at the inside face will be 1000°F. and the temperature at the outside face will be 120°F.?

*Solution.*—As a first approximation, assume that the temperature at the interface will be 600°F. Based on this assumption, the average temperatures of the inner and outer layers will be 800 and 360°F., and their thermal conductivities at these temperatures will be 0.059 and 0.044 B.t.u./(ft.)(hr.)(°F.), respectively. The actual outside diameter of a 2-in. pipe is 2.375 in. Hence, by Eq. (7), the rate of heat transfer per foot of length will be

$$q = \frac{2 \times 3.14 \times 1 \times (1000 - 120)}{2.3 \left[ \frac{\log_{10} (4.375/2.375)}{0.059} + \frac{\log_{10} (6.375/4.375)}{0.044} \right]}$$

$$\approx 292 \text{ B t u per hr}$$

Since this is also the rate of heat transfer through the individual layers, the temperature  $t$  at the interface can be calculated by Eq (4). Thus, using the data for the outer layer

$$292 = \frac{0.044 \times 2 \times 3.14 \times 1 \times (t - 120)}{2.3 \log_{10} (6.375/4.375)},$$

whence

$$t = 520^\circ\text{F}$$

Although a more accurate value could be obtained by repeating the calculations the foregoing value is sufficiently accurate to indicate that the 85% Magnesia will not be overheated and that the covering will therefore be satisfactory.

Equations (6) and (7) involve the assumption that no drop in temperature takes place at the boundary between the two materials. In practice, however, the contact between the layers is usually not perfect because of the roughness of the surfaces, and consequently a drop in temperature does take place. As a result, the actual rate of heat transfer by conduction is likely to be somewhat less than the calculated rate.

Equation (6) can be obtained as follows. Referring to Fig. 6, the rate of conduction through material  $a$  is, by Eq (3),

$$q = \frac{k_a A (\Delta t_a)}{L_a},$$

or

$$\Delta t_a = \frac{q L_a}{A k_a}$$

Similarly,

$$\Delta t_b = \frac{q L_b}{A k_b}$$

Since steady flow is assumed the rate of heat transfer  $q$  is the same through both materials. Hence, adding the last two equations,

$$t_1 - t_2 = \frac{q}{A} \left( \frac{L_a}{k_a} + \frac{L_b}{k_b} \right),$$

which is Eq (6).

Equation (7) is obtained from Eq (4) in a similar manner. An equation for calculating the rate of heat transfer by conduction

through two spherical bodies in series can be obtained from Eq. (5), but this case is of little practical importance.

**10. Unsteady Conduction of Heat through Flat Bodies.**—If the temperature at each surface of a homogeneous flat body remains uniform over the entire surface but changes with time, and if the temperatures within the body at some initial time are known, the temperatures within the body at any subsequent time can be

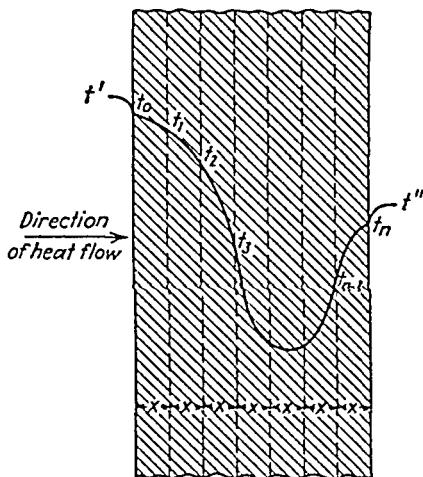


FIG. 8.—Temperature gradient for unsteady conduction through a homogeneous flat body.

closely approximated by the following method developed by E. Schmidt:<sup>1</sup>

1. The body is divided into  $n$  imaginary laminae of equal thicknesses  $x$ , as shown in Fig. 8. Any value may be chosen for  $n$ ; the larger the value the greater the accuracy of the method.

2. The temperatures  $t_0, t_1, t_2, \dots, t_n$  at the boundaries of these  $n$  imaginary laminae are to be determined at successive, equal intervals of time  $\theta$ , the value of  $\theta$  being calculated by the equation

$$\theta = \frac{x^2}{2} \left( \frac{c\rho}{k} \right), \quad (8)$$

<sup>1</sup> SCHMIDT, E., "A. Föppl's Festschrift," p. 179, J. Springer, Berlin, 1924.

where  $\theta$  = the time interval, hr

$x$  = the thickness of each imaginary lamina, ft

$\rho$  = the density of the material, lb per cu ft

$c$  = the specific heat of the material, B t u / (lb) (°F)

$k$  = the thermal conductivity of the material (assumed constant), B t u / (ft) (hr) (°F)

3 A table is prepared with the headings shown in Table I, the vertical columns being the temperatures  $t_0, t_1, t_2, \dots, t_n$ ,

TABLE I—CALCULATION FORM FOR UNSTEADY CONDUCTION THROUGH FLAT BODIES

Elapsed time hours	Temperatures -°F							
	$t_0$	$t_1$	$t_2$	$t_3$	$t_4$		$t_{n-1}$	$t_n$
0								
$\theta$								
$2\theta$								
$3\theta$								
$4\theta$								
$5\theta$								

and the horizontal rows being the values of these temperatures after the time intervals  $\theta, 2\theta, 3\theta$ , etc., have elapsed. The known initial temperatures are entered in the first horizontal row.

4 Each of the temperatures in the body of the table is equal to the arithmetic average of the two temperatures indicated by the arrows in Table I.

Case 1—If the surface temperatures  $t_0$  and  $t_n$  at the successive intervals of time  $\theta, 2\theta, 3\theta$ , etc., are known, the vertical columns of  $t_0$  and  $t_n$  can be filled in immediately and each of the temperatures in the body of the table can then be calculated.

Case 2—If the surface temperatures  $t_0$  and  $t_n$  of the body are not known but the temperatures  $t'$  and  $t''$  of the media in contact with the two surfaces are known, the values of  $t_0$  and  $t_n$  can be calculated by the equations



$$t_0 = \frac{h_i' x t' + k t_1}{k + h_i' x} \quad \text{and} \quad t_n = \frac{h_i'' x t'' + k t_{n-1}}{k + h_i'' x} \quad (9)$$

in which all the temperatures are evaluated for the same instant, and  $t_0$  and  $t_n$  = the surface temperatures of the body, °F.

$t_1$  and  $t_{n-1}$  = the temperatures within the body at distances  $x$  from the two surfaces, °F.

$t'$  and  $t''$  = the temperatures of the media in contact with the two surfaces, °F.

$h_i'$  and  $h_i''$  = the "combined coefficients" for convection and radiation from the surfaces of the body to the surrounding media (see Table 59 in Chap. VII), B.t.u./(sq. ft.)(hr.)(°F.).

$x$  = the thickness of the imaginary laminae, ft.

$k$  = the thermal conductivity of the material, B.t.u./(ft.)(hr.)(°F.).

After the temperatures existing at the boundaries of the imaginary laminae at successive intervals of time have been determined, the rates of heat transfer into or out of the body at any instant can be approximated by applying Eq. (3) to the outside laminae. The method of calculation is illustrated by the following problem:

**Illustrative Problem 7.**—An industrial furnace is to be operated 5 days a week but is to be shut off and allowed to cool down over the week end. It is to be electrically heated and is to operate at a temperature of 1800°F. Hence, the walls, which are to be 9 in. thick, may be constructed either of firebrick or of calcined, diatomaceous silica insulating brick. It is estimated that, in starting up,  $2\frac{1}{2}$  hr. will be required to raise the temperature of the inside surface of the wall from its initial temperature of 80°F., the ambient air temperature, to its operating temperature of 1800°F. In order to determine which construction will be the more economical (*i.e.*, in order to determine for which material the fixed charges plus the operating charges will be the smaller), the total heat loss from each square foot of surface per 120-hr. week is to be calculated for each material.

**Data.**—Evaluating the thermal conductivities of the two materials at an assumed average wall temperature of 1000°F., for the firebrick  $k = 0.67$  B.t.u./(ft.)(hr.)(°F.),  $c = 0.29$  B.t.u./(lb.)(°F.), and  $\rho = 125$  lb. per cu. ft.; and for the insulating brick  $k = 0.16$  B.t.u./(ft.)(hr.)(°F.),  $c = 0.23$  B.t.u./(lb.)(°F.), and  $\rho = 38$  lb. per cu. ft. The combined coefficient  $h_i''$  for convection and radiation from the outside surface of the wall to the surrounding air will increase as the temperature of the surface of the wall increases, but an average value of 3.0 B.t.u./(sq. ft.)(hr.)(°F.) may be assumed.

**Solution.**—If the wall is divided into  $n =$  four imaginary laminae,  $s = \frac{1}{2} \div 4 = 0.1875$  ft. Considering first the firebrick, by Eq. (8),

$$\theta = \frac{(0.1875)^2 \times 0.29 \times 125}{2 \times 0.67} \\ = 0.95 \text{ hr}$$

Assuming that the temperature  $t_2$  of the inside surface of the wall rises at a uniform rate during the first  $2\frac{1}{2}$  hr of operation, the values of  $t_2$  at the end of 0.95 and 1.90 hr can be determined by interpolation. Hence, the top horizontal row and the first vertical row of temperatures in Table 2 can be

TABLE 2—TEMPERATURE GRADIENTS AND RATES OF HEAT LOSS THROUGH FIREBRICK WALL

Elapsed time, hr	Temperature, °F					Rate of heat loss $q$ , Btu/(sq ft) (hr)
	$t_2$	$t_1$	$t_3$	$t_4$	$t_5$	
0	80	80	80	80	80	0
0.95	734	80	80	80	80	2340
1.90	1388	407	80	80	80	3510
2.85	1800	731	244	80	80	3810
3.80	1800	1022	407	162	80	2780
4.75	1800	1104	592	244	169	2490
5.70	1800	1196	674	381	243	2160
6.65	1800	1237	789	459	286	2010
7.60	1800	1295	848	538	328	1803
8.55	1800	1324	917	588	356	1700
9.50	1800	1359	956	637	382	1575
$\infty$	1800				475	1185

filled in immediately. All the temperatures for the first 4.75 hr with the exception of the last value of  $t_4$  can then be calculated by finding the arithmetic average of the two preceding temperatures indicated in Table 1. The last value of  $t_4$  can be calculated by Eq (9). Thus,

$$t_4 = \frac{3.0 \times 0.1875 \times 80 + 0.67 \times 244}{0.67 + 3.0 \times 0.1875} \\ = 169^\circ\text{F}$$

All the temperatures in the next horizontal row, with the exception of  $t_2$ , can then be calculated by averaging the preceding temperatures, and  $t_2$  can again be calculated by Eq (9). Proceeding in this way, all the temperatures in the rest of the table can be determined.

The rate of heat loss  $q$  from the interior of the furnace can be calculated by applying Eq (3) to the inside lamina. For example, the rate of heat loss from each square foot of surface after the elapse of 3.80 hr is

$$q = \frac{0.67 \times 1 \times (1800 - 1022)}{0.1875} \\ = 2780 \text{ Btu per hr}$$

After the wall reaches a state of equilibrium and steady conduction is established, the rate at which heat is transferred through the wall by conduction is equal to the rate at which heat is lost from the surface of the wall to the surrounding air. Thus, for each square foot of surface,

$$q = \frac{0.67 \times 1 \times (1800 - t_4)}{\frac{9}{12}} = 3.0 \times 1 \times (t_4 - 80).$$

Hence,  $t_4 = 475^\circ\text{F.}$ , and  $q = 1185$  B.t.u. per hr.

The values given in Table 3 for the insulating brick are calculated in exactly the same manner.

If the rates of heat loss  $q$  are plotted as ordinates against time as the abscissa, the area under the curve obtained is equal to the total heat loss.

TABLE 3.—TEMPERATURE GRADIENTS AND RATES OF HEAT LOSS THROUGH INSULATING BRICK WALL

Elapsed time, hr.	Temperature, °F.					Rate of heat loss $q$ , B.t.u./sq. ft.) (hr.)
	$t_0$	$t_1$	$t_2$	$t_3$	$t_4$	
0	80	80	80	80	80	0
0.96	747	80	80	80	80	570
1.92	1415	414	80	80	80	855
2.88	1800	748	247	80	80	899
3.84	1800	1024	414	164	80	663
4.80	1800	1107	594	247	117	592
5.76	1800	1197	677	356	141	515
6.72	1800	1239	777	409	152	479
7.68	1800	1289	824	465	164	436
8.64	1800	1312	877	494	171	417
9.60	1800	1339	903	524	177	393
$\infty$	1800	....	...	...	193	342

This is found to be 154,700 B.t.u. per sq. ft. per week for the firebrick and 43,300 B.t.u. per sq. ft. per week for the insulating brick.

It is interesting to note that, with the exception of approximately the first half hour of operation, the rate of heat loss from the interior of the furnace is always greater than the rate eventually reached when steady conduction is established. It may also be noted that, whereas the heat loss through the firebrick wall for steady conduction can be reduced by adding insulation to the outside surface of the wall, this insulation would have no effect whatsoever on the heat loss from the interior of the furnace during the first 6.65 hr. of operation.

Other simplified methods of solving unsteady-flow problems have been developed.<sup>1</sup> The advantage of the method given here

<sup>1</sup> *Ind. Eng. Chem.*, vol. 15, p. 1173, 1923. See also MARGARET FISHENDEN and O. A. SAUNDERS, "Calculation of Heat Transmission," His Majesty's

is the fact that it is applicable regardless of the manner in which the surface temperatures of the body may vary or what the initial temperature gradient may be. It has the disadvantage of being somewhat laborious if either too large a value is chosen for  $n$  or the temperature gradient after a long period of elapsed time is desired.

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Stationery Office (London), 1932 or A. SCHACK, 'Industrial Heat Transfer' translated by H. Goldschmidt and E. P. Partridge, John Wiley & Sons Inc., 1933

## CHAPTER II

### RADIATION

**11. Introduction.**—Radiant energy is emitted by all matter, the rate of emission depending upon the temperature and the nature of the matter. Unlike heat energy, radiant energy does not require the presence of any matter for its transmission. Thus, the earth receives radiant energy from the sun even though separated from it by an almost perfect vacuum. On encountering any matter, radiant energy may be partly absorbed by the matter and converted into heat energy; it may be partly reflected by the matter; or it may be transmitted through the matter. Consider, for example, a bare steam pipe running through a room. The radiation emitted by the pipe is almost all transmitted through the air in the room but is either absorbed or reflected by the walls. The walls, on the other hand, also emit radiation, part of which falls upon the pipe and is partly absorbed. The net rate at which the pipe loses heat by radiation is equal to the rate at which it emits radiation, minus the rate at which it absorbs radiation from the surrounding walls. Thus, the net rate at which any solid body gains or loses heat by radiation depends not only upon the temperature and the nature of its own surface but also upon the temperature and the nature of the surrounding surfaces.

Regarding the effect of the nature of the surfaces, it has been found experimentally that different surfaces *all at the same temperature* emit radiant energy at different rates but that there is a maximum rate that none of them exceeds. It has also been found that different surfaces absorb different fractions of the radiant energy incident upon them. The hypothetical body whose surface would absorb all the radiation incident upon it is called a *black body*. It can be shown that such a body also has the important property of emitting radiation at the maximum possible rate at each temperature.

The problem of calculating the net rate of heat transfer by radiation between two bodies whose surfaces are at uniform

temperatures and are separated by some medium that absorbs only a negligible portion of the radiant energy passing through it<sup>1</sup> is treated in this chapter. A method of solving such problems is explained in the next section, and the theory upon which this method is based is discussed in the remaining sections of the chapter.

## 12 Calculation of Heat Transfer by Radiation *a Emissivity*

In order to calculate the net rate at which any body gains or loses heat by radiation, it is necessary to know the *emissivity* of its surface. This quantity may be defined either as the ratio of the rate at which the given surface emits radiation to the rate at which a black body at the same temperature would emit radiation or as the ratio of the rate at which the given surface absorbs the radiation incident upon it to the rate at which a black body would absorb the same radiation, these two ratios being equal<sup>2</sup>. Thus, the emissivity of a surface may be thought of as its "efficiency" so far as its ability to emit or absorb radiation is concerned. The emissivity of most surfaces is not constant but increases with increase in temperature. Values of the emissivity for a number of surfaces are listed in Table IV in the Appendix.



Fig 9 —  
Radiation from  
a grooved sur-  
face

*b Effective Area*—The rate at which any body gains or loses heat by radiation depends also upon the shape of its surface. Consider for example, a body having a plane surface traversed by a groove, as shown in Fig 9. Assuming uniform temperature, each element of the surface emits radiation at the same rate, but since part of the radiation from each element of the surface of the groove is intercepted and absorbed by the walls of the groove, less heat is lost from the body per unit area of the grooved surface than per unit area of the plane surface. Hence,

<sup>1</sup> Most gases absorb very little radiant energy although water vapor, carbon dioxide, sulfur dioxide, methane, and ammonia are important exceptions. Charts for calculating the rate of heat transfer by radiation between these gases and the surfaces of solids have been prepared by Prof. H. C. Hottel. See *Trans. Am. Inst. Chem. Eng.*, vol. 19, 1927, and *Ind. Eng. Chem.*, vol. 19, p. 888, 1927.

<sup>2</sup> This definition is exact only if the surfaces are "gray" as explained in Sec. 18.

the effective area of any surface is less than the actual area if the surface contains any concave portions. The *effective area* may be assumed equal to the area of the surface that would be obtained by filling up all the concave portions of the actual surface to make them flush with the adjacent plane or convex portions; and the emissivity of this imaginary surface may be

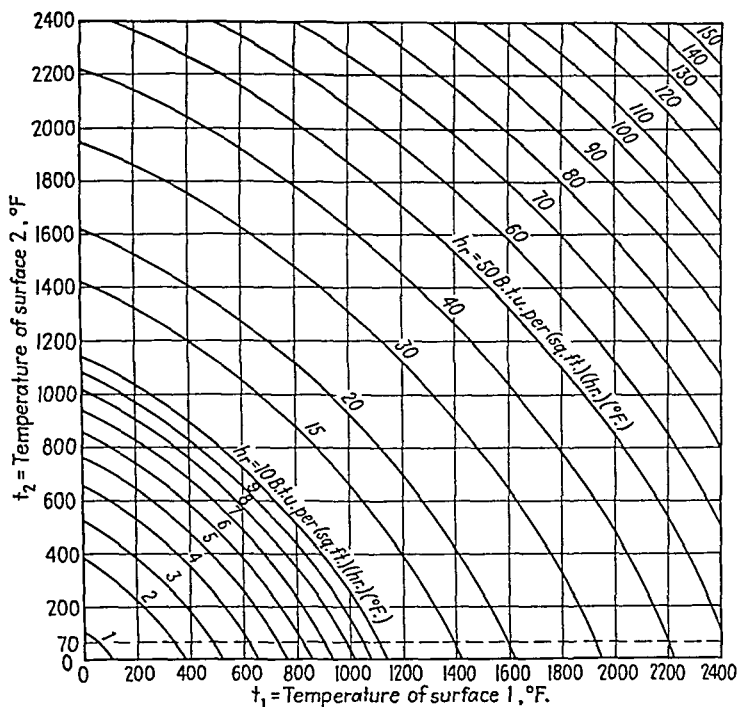


FIG. 10.—Radiation coefficient  $h_r$ , calculated by Eq. (21) and used in Eqs. (10) and (11).

taken as that of the actual surface increased toward unity in proportion to the depth and number of indentations.

*c. Equations for Calculating the Net Rate of Heat Transfer by Radiation between Two Surfaces.*—If two surfaces 1 and 2 are each uniform in temperature and are separated by a nonabsorbing medium (such as dry air), the net rate at which heat is transferred by radiation between the two surfaces can be calculated by the following equations:

**Case 1**—If all the radiation from surface 1 is intercepted by surface 2 (e.g., if surface 1 is completely inclosed by surface 2, or if surface 1 is flat and surface 2 extends as far as the plane of surface 1),

$$q = h_r A_1 (t_1 - t_2) F_s \quad (10)$$

where  $q$  = the net rate at which heat is transferred from surface 1 to surface 2 by radiation, B t u per hr

$h_r$  = the radiation coefficient of heat transfer, determined from Fig 10 or calculated by Eq (21), B t u / (sq ft) (hr) (°F)

$A_1$  = the effective area of surface 1, calculated as explained in the preceding paragraph, sq ft

$t_1$  and  $t_2$  = the temperatures of surfaces 1 and 2, °F

$F_s$  = a factor to allow for the emissivities of the two surfaces, determined from Table 4

TABLE 4—EMISSIVITY FACTOR  $F_s$

$e_1$  and  $e_2$  denote the emissivities of surfaces 1 and 2 and are evaluated from Table IV in the Appendix

No	Type of surface	$F_s$
1	Surface 1 the smaller of two concentric cylinders of radii $r_1$ and $r_2$ and of infinite length	$\frac{1}{\frac{1}{e_1} + \frac{r_1}{r_2} \left( \frac{1}{e_2} - 1 \right)}$
2	Surface 1 the smaller of two concentric spheres of radii $r_1$ and $r_2$	$\frac{1}{\frac{1}{e_1} + \left( \frac{r_1}{r_2} \right)^2 \left( \frac{1}{e_2} - 1 \right)}$
3	Surface 1 either of two infinite parallel planes	$\frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$
4	Surface 1 of any shape but small compared to surface 2	$e_1$
5	Surface 1 of any shape but almost as large as surface 2	$\frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$
6	Intermediate case between (4) and (5) above	$e_1 > F_s > \frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$

**Case 2**—If only part of the radiation from surface 1 is intercepted by surface 2 (i.e., surface 2 does not completely inclose surface 1),

$$q = h_r e_1 e_2 A_1 (t_1 - t_2) F_s \quad (11)$$

where  $q$ ,  $h_r$ ,  $A_1$ ,  $t_1$ , and  $t_2$  have the same meanings as in Eq (10), and



$e_1$  and  $e_2$  = the emissivities of surfaces 1 and 2, evaluated from Table IV in the Appendix.

$F_A$  = the fraction of the total radiation from surface 1 which falls upon surface 2, determined for several types of surfaces from Figs. 11 and 12.<sup>1</sup>

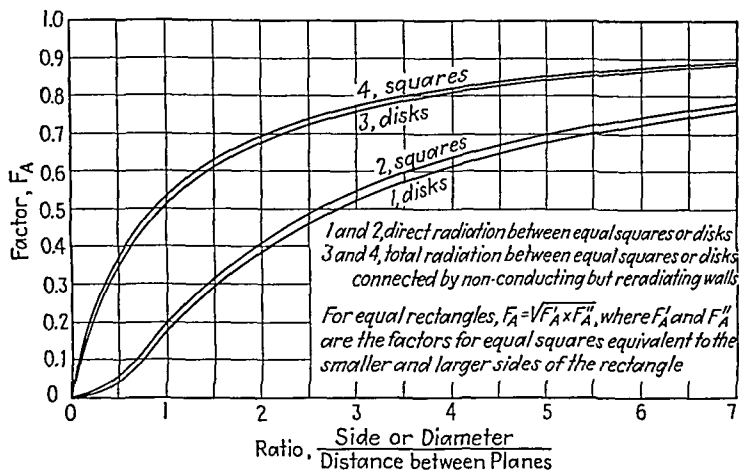


FIG. 11.—Radiation between equal squares or disks in parallel planes, directly opposed. (H. C. Hottel.)

This equation is not exact, but the approximation is satisfactory if  $e_1$  and  $e_2$  are 0.8 or higher.

It should be noted that the value of  $q$  calculated by Eq. (10) represents the net rate at which surface 1 gains or loses heat by radiation but that the value of  $q$  calculated by Eq. (11) represents only the net rate at which surface 1 gains or loses heat by radiation to surface 2 and does not include the heat gained or lost by radiation to other surfaces.

**Illustrative Problem 1.**—Calculate the rate at which heat is lost by radiation from a bare 3-in. nominal diameter wrought-iron pipe at 300°F. running through a room whose walls are at 70°F.

**Solution.**—Since the pipe is totally inclosed by the walls of the room, this problem is included in Case 1. From Fig. 10,  $h_r = 1.9$  B.t.u./(sq. ft.)(hr.)(°F.); and from Table 4, type 4,  $F_s = e_1$ . From Table IV in the Appendix, the emissivity  $e_1$  of dull, oxidized wrought iron is 0.94. Since

<sup>1</sup> Taken from Radiant Heat Transmission by H. C. Hottel, *Mech. Eng.*, vol. 52, p. 699, 1930.

the actual outside diameter of a 3-in. pipe is 3.500 in., the heat loss due to radiation per foot of length is, by Eq. (10),

$$q = 1.9 \times 3.14 \times \frac{3.500}{12} \times (300 - 70) \times 0.94 \\ = 376 \text{ B t u per hr}$$

**Illustrative Problem 2**—Liquid oxygen is to be stored at atmospheric pressure in a metal Dewar flask consisting of a 12-in. diameter inner sphere inclosed in a 13-in. diameter outer sphere. The spheres are made of brass, and the surfaces forming the vacuum space are highly polished. Calculate

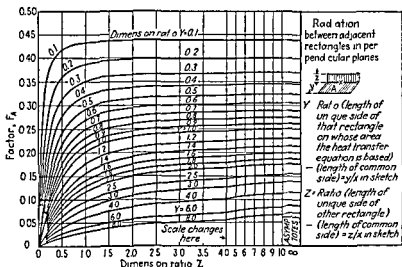


FIG. 12—Radiation between adjacent rectangles in perpendicular planes (H. C. Hotell)

the rate at which the oxygen will be lost by evaporation if the temperature of the outer sphere will be  $50^\circ\text{F}$  ( $510^\circ\text{F}$  abs.), the temperature of the inner sphere will be  $-298^\circ\text{F}$  ( $162^\circ\text{F}$  abs.), and the heat of vaporization of liquid oxygen is 91.8 B t u per lb

**Solution**—This problem is included in Case 1. By Eq. (21),

$$h_r = \frac{0.173 \times 10^{-8} \times (162^4 - 510^4)}{(-298 - 50)} \\ = 0.332 \text{ B t u / (sq ft) (hr) } (^\circ\text{F})$$

The surface area of the inner sphere is 3.14 sq ft, and the factor  $F_a$  is obtained from Table 4, type 2. The emissivities of highly polished brass at the temperatures involved in the present problem are not given in Table IV in the Appendix. However, a conservative result should be obtained if a constant value of 0.028 is assumed. Hence, by Eq. (10),

$$q = 0.332 \times 3.14 \times (-298 - 50) \left[ \frac{1}{\frac{1}{0.028} + \left(\frac{12}{13}\right)^2 \left(\frac{1}{0.028} - 1\right)} \right]$$

$$= -5.56 \text{ B.t.u. per hr.,}$$

the negative sign indicating that the direction of heat flow will be from the outer to the inner sphere. Assuming the heat transfer by convection to be negligible, the rate at which oxygen will be lost by evaporation is

$$\frac{5.56}{91.8} = 0.061 \text{ lb. per hr.}$$

**Illustrative Problem 3.**—Calculate the rate at which heat is lost by radiation through a 3-in. square peephole in a furnace wall 6 in. thick, the temperature within the furnace being 2200°F. and the surroundings being at 70°F.

*Solution.*—Since all the radiation entering the peephole from within the furnace must pass through an imaginary plane surface covering the inside end of the peephole, and since all the radiation lost through the peephole must pass through an imaginary plane surface covering the outside end, the problem is equivalent to one of calculating the rate of heat transfer by radiation between two parallel 3-in. squares 6 in. apart, one square being at 2200°F. and the other at 70°F. This problem is included in Case 2.

For  $t_1 = 2200^\circ\text{F.}$  and  $t_2 = 70^\circ\text{F.}$ , from Fig. 10,  $h_r = 40 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.})$ . Since none of the radiation falling upon either of the squares is reflected, these imaginary surfaces are equivalent to black bodies and have emissivities equal to unity. Assuming that the walls of the peephole reradiate all the radiation falling upon them, from Fig. 11, curve 4,  $F_A = 0.38$ . Hence, the rate of heat loss due to radiation through the peephole is, by Eq. (11),

$$q = 40 \times 1 \times 1 \times \frac{3 \times 3}{144} \times (2200 - 70) \times 0.38$$

$$= 2020 \text{ B.t.u. per hr.}$$

**Illustrative Problem 4.**—A muffle-type furnace is 6 ft. wide, 12 ft. deep, and 6 ft. high. What fraction of the radiation emitted by the floor of the furnace falls (1) upon the side walls and (2) upon the top of the furnace? Provided they are well insulated, the side walls will reradiate all the heat that they receive after they have reached a state of equilibrium. (3) Under these conditions, what fraction of the heat emitted by the floor of the furnace will reach the top of the furnace?

*Solution.*—1. Referring to Fig. 12, for each of the 6- by 12-ft. side walls,  $Y = \frac{1}{2}$  and  $Z = \frac{1}{2}$ , and therefore  $F_A = 0.24$ . For each of the 6- by 6-ft. side walls,  $Y = \frac{1}{2}$  and  $Z = \frac{1}{2}$ , and therefore  $F_A = 0.12$ . Hence, the four side walls receive  $2 \times (0.24 + 0.12) = 0.72$  of the radiation emitted by the floor.

2. Referring to Fig. 11, curve 2,  $F_A'$  for two 6-ft. squares 6 ft. apart is 0.20, and  $F_A''$  for two 12-ft. squares 6 ft. apart is 0.41. Hence, the fraction  $F_A$  of the radiation emitted by the floor and received by the top of the

length at any given temperature to the maximum possible rate at which radiant energy of the same wave length can be emitted by any surface at the same temperature. This second definition follows from the first, since  $I d\lambda$  is equal to the rate  $dq$  at which any surface emits radiant energy of wavelengths from  $\lambda$  to  $\lambda + d\lambda$ .

The *absorptivity*  $a$  of a surface for radiation of any wave length is the fraction of the incident radiation of the given wave length that the surface absorbs.

**15. Kirchhoff's Law**—Consider any two isolated, parallel plane surfaces that are both at the same temperature and that transmit none of the radiation incident upon them. Let  $q_\lambda$  denote the maximum possible rate per unit area at which radiation of wave length  $\lambda$  can be emitted by any surface at the given temperature and let  $e_1$ ,  $e_2$  and  $a_1$ ,  $a_2$  denote the emissivities and absorptivities of the two surfaces for radiation of this wave length. The rate per unit area at which each surface emits radiation of wave length  $\lambda$  is equal to  $e q_\lambda$ . Since the surfaces are parallel, all this radiation reaches the other surface, where the fraction  $a$  is absorbed<sup>1</sup> and the fraction  $(1 - a)$  is reflected. The reflected radiation all returns to the first surface, and again the fraction  $a$  is absorbed, and the fraction  $(1 - a)$  is reflected. Thus, part of the radiation emitted by each surface undergoes repeated reflections, the fraction  $a$  being absorbed and the fraction  $(1 - a)$  being reflected each time that the radiation falls upon either of the surfaces.

Since the two surfaces are isolated and are at the same temperature, the net rate at which each loses heat by radiation is zero. Hence, the rate  $e q_\lambda$  at which each surface emits radiation of wave length  $\lambda$  is equal to the sum of (1) the rate at which it absorbs the radiation of wave length  $\lambda$  emitted by the other surface plus (2) the rate at which it absorbs the radiation of wave length  $\lambda$  returned to it by reflection. These two equations are readily reduced to the single equation

$$\frac{e_1}{a_1} = \frac{e_2}{a_2} \quad (12)$$

<sup>1</sup>The emissivities and absorptivities of the two surfaces are assumed independent of the angles of emission and incidence. The assumption that the emissivity is independent of the angle of emission is known as *Lambert's cosine principle*.

This result is known as *Kirchhoff's law* and may be stated, "For any given wave length and temperature, the ratio of the emissivity to the absorptivity is a constant for all surfaces."

**16. Black Body.**—A *black body* is defined as one that absorbs all the radiation of all wave lengths falling upon it. From this definition it follows that the absorptivity  $a$  of a black body is unity at all wave lengths and temperatures. No actual surfaces are black in this sense, although the imaginary surface covering a small opening in an otherwise completely enclosed box is approximately so, since all the radiation falling upon it enters the box and undergoes successive reflections until completely absorbed.

**17. Equivalence of Emissivity and Absorptivity.**—Kirchhoff's law states in effect that any surface emits radiation in proportion to its ability to absorb radiation. It follows that a black body, which absorbs a maximum fraction of the radiation of all wave lengths incident upon it (*i.e.*, all of it), must also emit radiation of all wave lengths at the maximum possible rate. Hence, from the second definition of emissivity given in Sec. 14, it follows that the emissivity  $e$  of a black body is unity at all wave lengths. Since the emissivity and the absorptivity of a black body are equal at all wave lengths and temperatures, it follows from Kirchhoff's law that *the emissivity and the absorptivity of any surface are equal at the same wave length and temperature*. Thus, at the same wave length and temperature, for any surface,

$$e = a. \quad (13)$$

**18. Gray Surfaces.**—A *gray surface* is defined as one whose emissivity  $e$  is the same at all wave lengths and temperatures. Hence, the ratio of the rate at which a gray surface emits radiation of any wave length to the rate at which a black body at the same temperature would emit radiation of the given wave length is constant for all wave lengths. Therefore, *for a gray surface*,

$$e = (\text{the total rate at which a gray surface emits radiation of all wave lengths}) \div (\text{the total rate at which a black body at the same temperature would emit radiation of all wave lengths}).^1 \quad (14)$$

<sup>1</sup> Applied to any actual surface, the quantity defined in this way is called the *total*, or *hemispherical*, *emissivity*.

By Eq (13) the absorptivity  $\alpha$  of a gray surface is also the same at all wave lengths and temperatures. Therefore, *for a gray surface,*

$$\alpha = \frac{\text{(the total rate at which a gray surface absorbs the radiation of all wave lengths incident upon it)}}{\text{(the total rate at which a black body would absorb the same radiation)}} \quad (15)$$

The definition of emissivity given in Sec 12a is based on Eqs (13), (14) and (15) and is therefore strictly valid only for gray surfaces. None of the surfaces encountered in engineering work is actually "gray," but the error introduced by assuming them to be such is usually small.

**19 Stefan-Boltzmann Equation**—The total rate at which a black body at any temperature emits radiation of all wave lengths is represented by the area under the intensity wave-length curve for a black body (Fig 13) and can be calculated by the Stefan Boltzmann equation

$$q = 0.173 \times 10^{-8} AT^4, \quad (16)$$

where  $q$  = the total rate at which a black body emits radiation of all wave lengths B t u per hr

$A$  = the area of the black body sq ft

$T$  = the absolute temperature of the black body, °F abs

This equation was first established by Stefan on the basis of the experimental work of Dulong and Petit and was subsequently deduced by Boltzmann from purely theoretical considerations.

It follows from Eqs (14) and (16) that the total rate at which a gray surface at any temperature emits radiation of all wave lengths is given by the equation

$$q = 0.173 \times 10^{-8} \epsilon AT^4 \quad (17)$$

where  $\epsilon$  represents the emissivity of the gray surface.

**20 Net Rate of Heat Transfer by Radiation between Two Surfaces**—If the surfaces are separated by some nonabsorbing medium the net rate at which any surface 1 loses heat by radiation to some other surface 2 is equal to (1) the rate at which surface 1 emits radiation (2) minus the rate at which it reabsorbs the fraction of this radiation that is reflected back to it by surface 2, and (3) minus the rate at which it absorbs the radiation emitted by surface 2.

*Case 1.*—Consider first the case where all the radiation emitted by surface 1 falls upon surface 2. The simplest problem of this type is that of a gray surface at the uniform absolute temperature  $T_1$  completely inclosed by a black body at the uniform absolute temperature  $T_2$ . Letting  $A_1$  denote the area and  $e_1$  the emissivity of the gray surface, by Eq. (17) the rate at which the gray surface emits radiation is  $0.173 \times 10^{-8} e_1 A_1 T_1^4$ . Since all this radiation is absorbed by the black body, item (2) in the preceding paragraph is zero. The rate at which the gray surface absorbs the radiation emitted by the black body can be determined as follows: If the gray surface were replaced by a black body of the same size as the gray surface but at the temperature  $T_2$ , such a body would be in equilibrium with its surroundings and would therefore emit and absorb radiation at the same rate. This rate, by Eq. (16), is equal to  $0.173 \times 10^{-8} A_1 T_2^4$  and is the rate at which radiation falls upon the gray surface. Since absorptivity and emissivity are numerically equal, by Eq. (15) the rate at which the gray surface absorbs radiation is

$$0.173 \times 10^{-8} e_1 A_1 T_2^4.$$

Hence, the net rate  $q$  at which the gray surface loses heat by radiation is

$$q = 0.173 \times 10^{-8} e_1 A_1 (T_1^4 - T_2^4). \quad (18)$$

If surface 2 is gray instead of black, part of the radiation that reaches it from surface 1 is reflected and returned to surface 1. Hence, the net rate at which surface 1 loses heat by radiation depends not only upon the emissivities of both surfaces but also upon the shapes and relative positions of the two surfaces. Assuming that the reflection is diffuse rather than specular, it can be shown<sup>1</sup> that the net rate at which the smaller of two concentric cylinders of infinite length loses heat by radiation is

$$q = 0.173 \times 10^{-8} A_1 \left[ \frac{1}{\frac{1}{e_1} + \frac{r_1}{r_2} \left( \frac{1}{e_2} - 1 \right)} \right] (T_1^4 - T_2^4), \quad (19)$$

and the net rate at which the smaller of two concentric spheres loses heat by radiation is

$$q = 0.173 \times 10^{-8} A_1 \left[ \frac{1}{\frac{1}{e_1} + \left( \frac{r_1}{r_2} \right)^2 \left( \frac{1}{e_2} - 1 \right)} \right] (T_1^4 - T_2^4), \quad (20)$$

<sup>1</sup> *Proc. Phys. Soc.*, vol. 41, p. 569, 1929.

where the subscript 1 indicates the smaller of the two cylinders or spheres  $r$  denotes radius and the other symbols have the same meanings as in Eq (18). The term in brackets in these two equations is the factor  $F_r$  listed in Table 4 the last four cases in this table being special cases of the first two.

The radiation coefficient of heat transfer  $h_r$  given in Fig 10 is defined by the equation

$$h_r = \frac{0.173 \times 10^{-8}(T_1^4 - T_2^4)}{(t_1 - t_2)} \quad (21)$$

Equation (10) is obtained by combining Eq (21) with either Eq (19) or Eq (20) and replacing the term in brackets with  $F_r$ .

*Case 2*—Consider next the case where only part of the radiation emitted by surface 1 falls upon surface 2. Letting  $f$  denote the fraction of the radiation that is emitted by an element  $dA$  of surface 1 and that falls upon surface 2 and remembering that surface 2 absorbs the fraction  $e_2$  of the radiation falling on it the rate at which surface 2 absorbs radiation from  $dA$  is  $0.173 \times 10^{-8}e_1e_2dAT_1^4f$ . The rate at which radiation from surface 2 falls upon  $dA$  is  $0.173 \times 10^{-8}e_2dAT_2^4f$  since this is the rate at which an element of surface 2 placed in the position of  $dA$  would emit radiation to or absorb radiation from surface 2. Hence the rate at which  $dA$  absorbs radiation from surface 2 is  $0.173 \times 10^{-8}e_1e_2dAT_2^4f$ . Neglecting the rates at which the radiation reflected back to each surface is reabsorbed (permissible if  $e_1$  and  $e_2$  are both large) the net rate at which heat is transferred from  $dA$  to surface 2 by radiation is

$$0.173 \times 10^{-8}e_1e_2dA(T_1^4 - T_2^4)f$$

and the net rate  $q$  at which heat is transferred from the entire surface 1 to surface 2 is

$$q = 0.173 \times 10^{-8}e_1e_2A_1(T_1^4 - T_2^4)F_A \quad (22)$$

where  $F_A$  is the average of the values of  $f$  for each element of surface 1. Equation (11) is obtained by combining this equation with Eq (21).



## CHAPTER III

### CONVECTION

21. **Introduction.**—Convection is the transfer of heat from one part of a fluid to another by the mixing of the warmer particles of the fluid with the cooler. The motion of the fluid may be caused either by differences in density within the fluid or by some mechanical device, such as a stirrer, pump, or fan. The first process is called *natural convection*; and the second, *forced convection*.

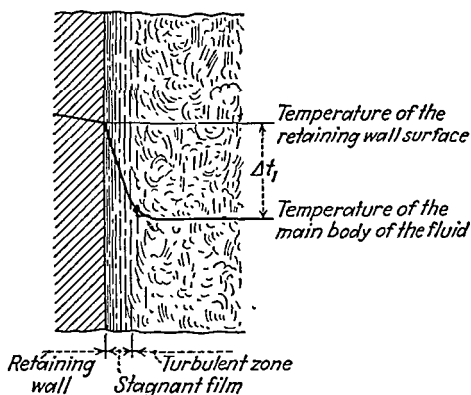


FIG. 14.—Temperature gradient through a fluid in contact with a warmer retaining wall.

Consider the case of a fluid in contact with a retaining wall whose temperature is higher than that of the fluid, as shown in Fig. 14. Although the main body of the fluid is likely to be in turbulent motion, the thin film of fluid in contact with the retaining wall is relatively stagnant as the result of friction with the wall. The thickness of this stagnant film is not clearly defined and varies as the degree of turbulence of the adjacent fluid varies. Since heat is transferred through this film largely by conduction rather than by convection, the entire process by which heat is transferred from the retaining wall to the main

body of the fluid is rather complicated. Calculations are simplified, however, by the use of so-called *convection coefficients* which include the combined effect of both conduction and convection.

In this and the following chapter, it is assumed that a state of equilibrium has been reached, i.e., the temperature gradient is assumed to remain constant and not change with time, and the rate of heat transfer through the retaining wall is assumed equal to the rate at which heat is absorbed or given up by the fluid in contact with the wall. In addition, it is assumed that radiation is either negligible or else is taken into account by the method given in Chap. II. In this connection, the rate of heat transfer by radiation is usually small compared to the rate of heat transfer by convection. An important exception is that of heat transfer by natural convection from any surface to a gas when the surface is exposed to other surfaces at higher or lower temperatures. For example, a bare steam pipe may lose more heat by radiation to the walls of a room than it loses by natural convection to the surrounding air.

**22 Definition of the Film Coefficient  $h$** —The *film coefficient* of convection  $h$  is defined as the rate of heat transfer between the retaining wall and the fluid per unit area of the retaining wall and per degree temperature difference between the surface of the wall and the main body of the fluid. From this definition it follows that

$$q = hA(\Delta t_1), \quad (23)$$

where  $q$  = the rate of heat transfer by convection, B t u per hr

$h$  = the film coefficient, B t u / (sq ft) (hr) (°F)

$A$  = the area of the retaining wall, sq ft

$\Delta t_1$  = the temperature difference between the surface of the wall and the main body of the fluid °F

The quantity  $h$  is also called the *individual coefficient* or the *surface coefficient*.

**23 Factors Affecting the Film Coefficient  $h$** —There are three types of heat transfer by convection. (1) heat transfer in which the fluid does not change phase (i.e., the fluid remains either a liquid or a gas throughout the process), (2) heat transfer in which a vapor is condensed and (3) heat transfer in which a liquid is evaporated. The film coefficients for all three types depend not only upon the kind of fluid involved and upon the pressure,

and temperature of the fluid but also upon the type of retaining wall surface (*e.g.*, flat surfaces, the inside surface of tubes, or the outside surface of tubes) and upon the position of the retaining wall surface (*e.g.*, horizontal or vertical). In addition, the film coefficients for each individual type of convection depend upon the following factors:

1. For the first type in which the fluid undergoes no change of phase, the film coefficient depends upon the turbulence of the fluid and increases as the turbulence increases. In the case of natural convection, the turbulence, and consequently the film coefficient, increases as the temperature difference between the main body of the fluid and the surface of the retaining wall increases. In the case of flow through tubes, regardless of whether the motion is the result of natural or forced convection, the film coefficient increases as the velocity of the fluid increases. Although the type of flow depends also upon the diameter of the tube and upon the density and viscosity of the fluid,<sup>1</sup> at low velocities the flow is likely to be *streamline* (also called *viscous* or *laminar* flow), and at higher velocities it is usually *turbulent*. The transition from streamline to turbulent flow is accompanied by a considerable increase in the film coefficient. In the case of flow outside tubes, the film coefficient depends also upon the direction of flow. For any given velocity, the film coefficient is usually higher for flow normal to the tubes than for flow parallel to the tubes.

The film coefficient depends also upon whether the fluid is being heated or cooled, because the temperature of the film of fluid in contact with the surface of the retaining wall is at a higher temperature than the main body of the fluid when the fluid is being heated and lower when cooled. The difference in the film coefficients for heating and for cooling decreases as the turbulence of the fluid increases and in the case of gases is usually negligible.

2. For the second type of convection in which a vapor is condensed, the film coefficient depends upon whether the condensation is *dropwise* or *filmwise*. As the names imply, with dropwise

<sup>1</sup> For isothermal flow inside straight tubes, for example, the flow is usually streamline if the product  $DV\rho/\mu$  [ $D$  = inside tube diameter, ft.;  $V$  = velocity, ft. per sec.;  $\rho$  = density, lb. per cu. ft.; and  $\mu$  = viscosity, lb./(ft.)(sec.)] is less than 2,100 and is turbulent, if greater.

condensation the condensate collects on the surface of the retaining wall in drops, with filmwise condensation, the condensate forms a continuous film over the entire surface. Although film coefficients for dropwise condensation are four to eight times as high as for filmwise, the latter type is the one more likely to occur in practice, since dropwise condensation usually takes place only on polished surfaces.

In the case of filmwise condensation, the film of condensate adhering to the surface of the retaining wall acts somewhat as insulation, and consequently the film coefficient depends upon both the thickness and the turbulence of this film. For example, if condensate is allowed to drip on to a horizontal tube from other tubes above it, the film of condensate will be thicker, and therefore the film coefficient will be smaller. Similarly, since all the condensate formed on a vertical surface must flow down the surface, the average thickness of the film of condensate tends to increase as the height of the surface is increased. However, up to a certain rate of condensation the flow of condensate down the surface is streamline, and for greater rates it is turbulent. As long as the flow is streamline, the average film coefficient decreases as the height of the surface is increased, but if the flow is turbulent, both the turbulence and the film coefficient increase as the height is increased.

The film coefficient is also affected by the velocity of the vapor. If the vapor strikes the surface of the retaining wall with a velocity sufficient to sweep away part of the film of condensate, the thickness of the film is reduced and the film coefficient is increased.

3 The various factors affecting the film coefficient for the third type of convection in which a liquid is evaporated are not yet so well established as are the factors affecting the film coefficients for the other two types. It is known, however, that the film coefficient depends largely upon how great a fraction of the heating surface is in contact with the liquid rather than with the vapor bubbles. Since the rate of heat transfer from the heating surface to the liquid is much higher than the rate of heat transfer to the vapor the film coefficient is increased by anything that either reduces the size of the vapor bubbles formed on the heating surface or reduces the time required for the bubbles to break away. The size of the bubbles is partially

determined by the tendency of the liquid to wet the heating surface. Thus, the film coefficient is higher for a liquid having a high tendency to wet the surface than for one having a low tendency. The time required for the bubbles to break away from the heating surface is greatly affected by the turbulence of the liquid. If the motion of the liquid is entirely the result of natural convection, both the turbulence and the film coefficient increase as the temperature difference between the heating surface and the main body of the fluid increases up to a certain critical value. As the temperature difference increases beyond this critical value, however, the film coefficient decreases very rapidly, because the heating surface becomes completely separated from the liquid by a film of vapor. If the liquid is violently agitated by some form of mechanical stirrer, the temperature difference has very little effect on the film coefficient. Finally, the film coefficient is affected by the condition of the heating surface and is higher if the surface is rough instead of smooth.

In addition to the various factors already mentioned, the film coefficients for all three types of heat transfer by convection are affected by the presence of scale or other deposits on the surface of the retaining wall. Such deposits act as insulation and may greatly reduce the rate of heat transfer. Allowance for them may be made by the method explained in Sec. 26.

Values of the film coefficient must be determined experimentally for most cases, because the motion of the fluid and the temperature distribution are too complicated to permit these coefficients to be predicted from purely theoretical considerations. Fortunately, the experimental results obtained with a few fluids can be used to predict film coefficients for other fluids by means of dimensional analysis (see Secs. 29 and 30). Dimensionless equations, suggested by dimensional analysis and representing with reasonable accuracy the results of numerous investigators, are available in the technical literature. Based on certain of these equations, charts and tables have been prepared which can be used to determine film coefficients for various fluids under various conditions. These charts and tables, together with the dimensionless equations upon which they are based, are given in Chap. IV.

✓ 24. **Definition of the Over-all Coefficient  $U$ .**—In most industrial heat-transfer equipment, one fluid receives heat from

another fluid, the two fluids being separated by a solid retaining wall. In designing such equipment, the temperatures at which the two fluids are to enter and leave the equipment are usually known, but the temperature of the retaining wall is not known. Consequently, the expected rate of heat transfer cannot be calculated directly from the film coefficients, since these are based on the temperature difference between the *retaining wall* and the fluids. However, the expected rate of heat transfer can be readily calculated from the over-all coefficient  $U$ , because this coefficient is based on the temperature difference between the two fluids.

The *over-all coefficient*  $U$  is defined as the rate at which heat is transferred from one fluid to another per unit area of the retaining wall and per degree temperature difference between the two fluids. From this definition, it follows that

$$q = UA(\Delta t), \quad (24)$$

where  $q$  = the rate of heat transfer by convection, B t u per hr

$U$  = the over-all coefficient, B t u / (sq ft) (hr) (°F)

$A$  = the area of the retaining wall, sq ft

$\Delta t$  = the temperature difference between the main body of one fluid and the main body of the other, °F

If the temperature difference between the two fluids is not constant throughout the equipment, an average value must be used for  $\Delta t$ . The correct method of calculating this average value is discussed in Sec. 27. If the heat transfer surface consists of tubes, the value of the area  $A$  to be used depends upon whether the coefficient  $U$  is based on the inside or outside area of the tubes, as explained in the next section.

**25 Relation between the Over-all Coefficient and the Film Coefficients**—The over-all coefficient  $U$  can be calculated from the film coefficients when these are known for both sides of the retaining wall.

For *flat* or slightly curved walls free from scale,

$$\frac{1}{U} = \frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}, \quad (25)$$

where  $U$  = the over all coefficient, B t u / (sq ft) (hr) (°F)

$h_1$  = the film coefficient for one side of the wall, B t u / (sq ft) (hr) (°F)

$h_2$  = the film coefficient for the other side of the wall,  
B.t.u./ (sq. ft.) (hr.) (°F.).

$L$  = the thickness of the wall, ft.

$k$  = the thermal conductivity of the wall, B.t.u./  
(ft.) (hr.) (°F.).

For tubes, the coefficient  $U$  may be based on either the inside or the outside area of the tubes. For ordinary thin-walled tubes free from scale, the over-all coefficient  $U$  may be calculated by the following approximate equations: Based on the inside area of the tube,

$$\frac{1}{U_1} = \frac{1}{h_1} + \frac{L}{k} + \frac{D_1}{h_2 D_2}, \quad (26)$$

and based on the outside area of the tube,

$$\frac{1}{U_2} = \frac{D_2}{h_1 D_1} + \frac{L}{k} + \frac{1}{h_2}, \quad (27)$$

where  $U_1$  and  $U_2$  = the over-all coefficients based on the inside and the outside tube areas, respectively,  
B.t.u./ (sq. ft.) (hr.) (°F.).

$h_1$  and  $h_2$  = the film coefficients for the inside and outside tube surfaces, respectively, B.t.u./  
(sq. ft.) (hr.) (°F.).

$D_1$  and  $D_2$  = the inside and outside diameters of the tube,  
respectively, ft. or in.

$L$  = the thickness of the tube wall, ft.

$k$  = the thermal conductivity of the tube wall,  
B.t.u./ (ft.) (hr.) (°F.).

For tubes having unusually heavy walls, the  $L/k$  terms in these two equations should be replaced by  $(D_1/2k) \log_e (D_2/D_1)$  and  $(D_2/2k) \log_e (D_2/D_1)$ , respectively.

✓ For practical purposes, sufficiently accurate values of the over-all coefficient  $U$  for both flat walls or thin-walled tubes free from scale can frequently be calculated by the following approximate equation:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}, \quad (28)$$

where  $h_1$  and  $h_2$  are the film coefficients for the two sides of the wall. This equation can be quickly solved by means of Fig. 15.

Equation (25) can be derived as follows. The rates of heat transfer between the two fluids and the surfaces of the retaining

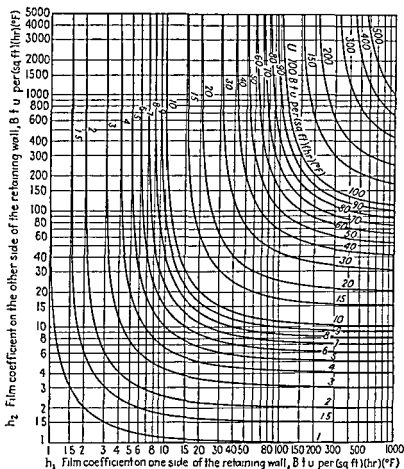


FIG. 15—Chart for determining approximate values of the over all coefficient  $U$  from the film coefficients  $h_1$  and  $h_2$  [based on Eq. (28)]

wall are, by Eq. (23),

$$q = h_1 A (\Delta t_1)$$

and

$$q = h_2 A (\Delta t_2),$$

where  $\Delta t_1$  and  $\Delta t_2$  are the temperature differences between the main body of each fluid and the surface of the wall. By Eq. (3)



in Chap. I the rate of heat transfer by conduction through the wall itself is

$$q = \frac{kA(\Delta t_w)}{L},$$

where  $\Delta t_w$  is the temperature difference between the two surfaces of the wall. By Eq. (24) the rate of heat transfer from one fluid to the other is

$$q = UA(\Delta t),$$

where  $\Delta t$  is the temperature difference between the main body of one fluid and the main body of the other. Since it is assumed that a state of equilibrium has been reached, the rate of heat transfer  $q$  is the same in all of these equations. Equation (25) is obtained by solving each of the equations for the temperature difference term and substituting these expressions into the equation

$$\Delta t = \Delta t_1 + \Delta t_w + \Delta t_2.$$

Equations (26) and (27) are obtained in a similar manner. These equations are based on the simplifying assumption that the rate of heat transfer through the tube wall itself can be calculated with sufficient accuracy by Eq. (3) in Chap. I. This assumption is permissible for ordinary thin-walled tubes, but for tubes having heavy walls the rate of heat transfer through the tube wall must be calculated by Eq. (4) in Chap. I.

Equation (28) is approximately equivalent to Eqs. (25), (26), and (27) if the retaining wall is thin and is made of some material having a high thermal conductivity, because for these conditions the  $L/k$  terms in these equations become negligible and the  $D_1/D_2$  ratio is approximately unity.

**26. Effect of Scale on the Over-all Coefficient  $U$ .**—The equations for calculating the over-all coefficient  $U$  given in the preceding section involve the assumption that the surfaces of the retaining wall are free from scale. If scale is present on either surface, the over-all coefficient  $U$  can be calculated by the equation

$$\frac{1}{U_s} = \frac{1}{U_c} + \frac{1}{h_s}, \quad (29)$$

where  $U_s$  = the over-all coefficient when one surface of the retaining wall is covered with scale, B.t.u./(sq. ft.)(hr.)(°F.).

$U_c$  = the over all coefficient when both surfaces of the retaining wall are clean, B t u / (sq ft) (hr) (°F)

$h_s$  = the scale coefficient B t u / (sq ft) (hr) (°F) which is defined as the rate of heat transfer through the scale per unit area of scaly surface and per degree temperature drop through the scale

If scale is present on both surfaces, a second  $1/h_s$  term must be added to the equation

Values of the scale coefficient  $h_s$  can be calculated if the thermal conductivity and the thickness of the scale are known In

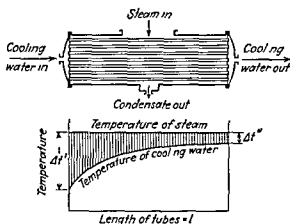


FIG. 16.—Variation of the temperature difference in a condenser

practice however it is more satisfactory to determine  $h_s$  from values of the over all coefficient  $U$  measured before and after the surfaces of the retaining wall have become covered with scale. A table of values of  $h_s$  is given in Chap. IV.

If a  $1/h_s$  term is added to the right-hand members of Eqs (25) (26) (27) and (28) these equations become equivalent to Eq (29). Equation (29) can be derived in a manner analogous to that used in deriving these equations.

**27 Mean Temperature Difference**—In general the temperature difference between two fluids flowing through any kind of heat-transfer equipment does not remain constant throughout the equipment. For example consider a steam condenser in which the cooling water makes a single pass through the tubes as shown in Fig. 16. The temperature difference between the steam and the cooling water is greatest at the section where the water enters

*exchangers* in which the two fluids flow at right angles to each other, the true mean temperature difference can be determined by

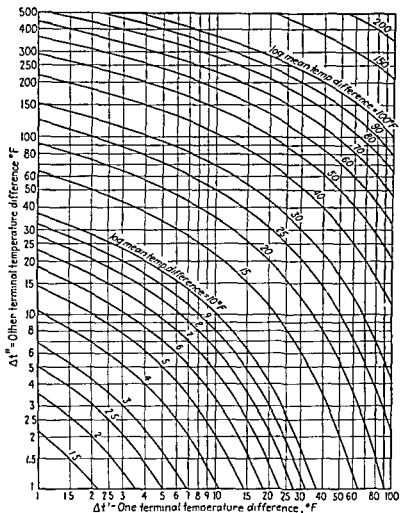


FIG. 17—Chart for determining the logarithmic mean temperature difference from the terminal temperature differences

multiplying the logarithmic-mean temperature difference calculated for counterflow by a correction factor  $F$  obtained from Fig 19, also prepared by Bowman, Mueller, and Nagle. The nomenclature used in these figures is defined in Table 5

If the temperature of one of the fluids remains constant, as shown in Fig. 16, Eq. (30) can be derived as follows: Let  $t$ ,  $c_p$ , and  $w$  denote the temperature, specific heat, and weight rate

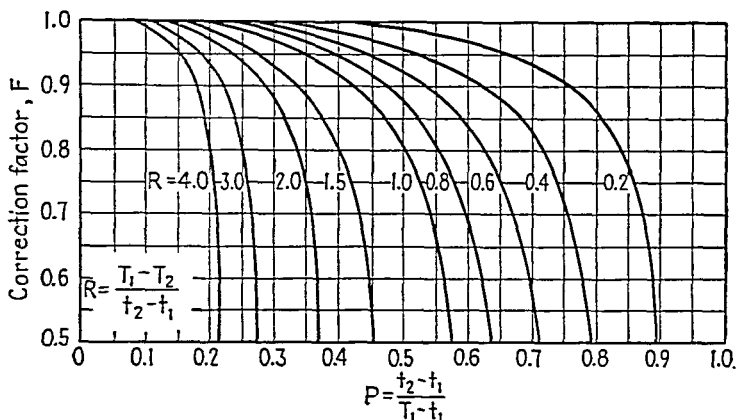


FIG. 18a.—Correction factor for multipass heat exchangers with one shell pass and two, four, or any multiple of tube passes. (Bowman, Mueller, and Nagle.)

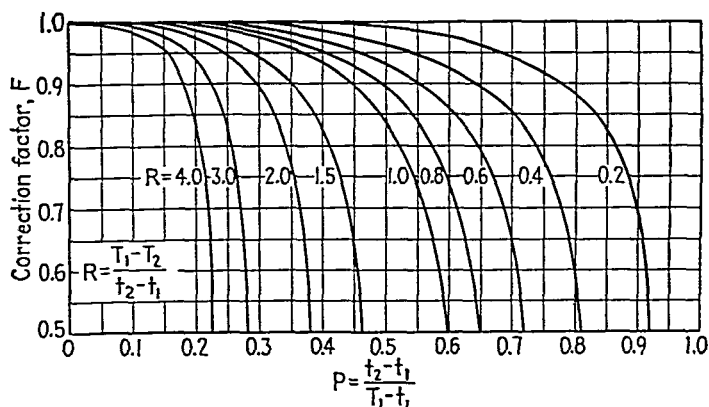


FIG. 18b.—Correction factor for multipass heat exchangers with one shell pass and three, six, or any multiple of tube passes, majority of tube passes in counterflow. (Bowman, Mueller, and Nagle.)

of flow, respectively, of the fluid whose temperature changes; let  $A'$  denote the total heat-transfer surface per foot of length of the tube bundle; and let  $l$  denote the length of the tube bundle. If the equipment is perfectly insulated, the rate of heat transfer

through a section of the tube bundle of length  $dl$  is equal to the heat absorbed per hour by the fluid flowing through this section

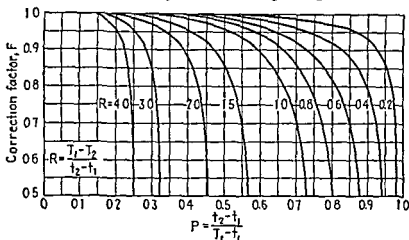


FIG 18c—Correction factor for multipass heat exchangers with two shell passes and four eight or any multiple of tube passes (Bowman Mueller and Nagle)

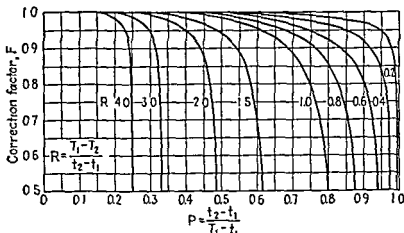


FIG 18d—Correction factor for multipass heat exchangers with three shell passes and six twelve or any multiple of tube passes (Bowman Mueller and Nagle)

Hence,

$$U(A'dl)(\Delta t) = wc_p dt = -uc_p d(\Delta t),$$

or

$$dl = -\frac{wc_p}{UA'} \frac{d(\Delta t)}{(\Delta t)}.$$

If the  $w c_p / UA'$  term remains constant, the following equation is obtained on integrating between the limits  $l = 0$ ,  $\Delta t = \Delta t'$  and  $l = l$ ,  $\Delta t = \Delta t''$ :

$$l = \frac{w c_p}{U A'} \cdot \ln \left( \frac{\Delta t'}{\Delta t''} \right).$$

The mean temperature difference for the entire tube bundle is

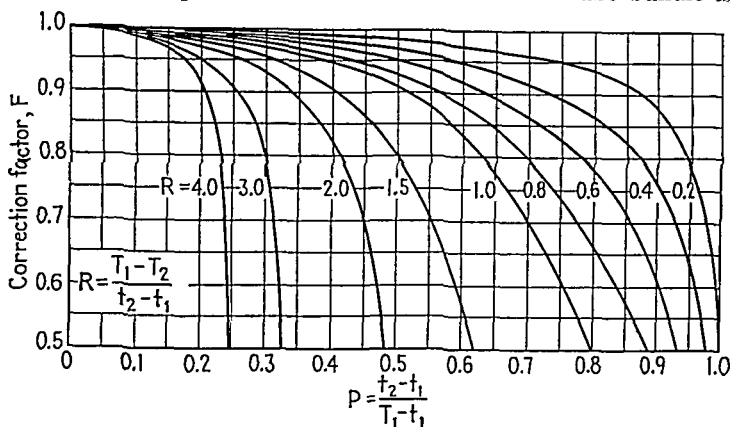


FIG. 19.—Correction factor for single-pass crossflow heat exchangers. (Bowman, Mueller, and Nagle.)

equal to the mean height of the shaded area in Fig. 16, or

$$\Delta t_{\text{mean}} = \frac{\text{shaded area}}{\text{length of shaded area}} = \frac{\int (\Delta t) dl}{l}.$$

Equation (30) is obtained by inserting the preceding expressions for  $l$  and  $dl$  into this equation and integrating between the limits  $\Delta t = \Delta t'$  and  $\Delta t = \Delta t''$ .

TABLE 5.—NOMENCLATURE FOR FIGS. 18 AND 19

$T_1$	Inlet temperature of shell-side fluid, °F.
$T_2$	Outlet temperature of shell-side fluid, °F.
$t_1$	Inlet temperature of tube-side fluid, °F.
$t_2$	Outlet temperature of tube-side fluid, °F.
$P$	Dimensionless ratio = $(t_2 - t_1) / (T_1 - t_1)$ .
$R$	Dimensionless ratio = $(T_1 - T_2) / (t_2 - t_1)$ .
$F$	Correction factor by which the logarithmic-mean temperature difference for counterflow is multiplied to obtain the true mean temperature difference.
$(T_1 - t_2)$	Terminal temperature difference for counterflow, °F.
$(T_2 - t_1)$	Terminal temperature difference for counterflow, °F.

It can be shown in a somewhat similar manner that Eq (30) also applies when the temperatures of both fluids vary, provided that the equipment is insulated and the quantity  $wc_p/UA'$  is constant for each fluid

**28 Design of Equipment Involving Heat Transfer by Convection**—The amount of heat transfer surface required in any equipment in which heat is to be transferred by convection from one fluid to another can be calculated by Eq (24)

$$q = UA(\Delta t) \quad (24)$$

It is first necessary to determine (1) the rate  $q$  at which heat is to be transferred from one fluid to the other, (2) the over-all coefficient of convection  $U$ , and (3) the mean temperature difference  $\Delta t$  between the two fluids. These three quantities can be calculated as follows

1 If the equipment is to be insulated, the rate of heat transfer  $q$  can be calculated from the properties of either fluid, since the rate at which the warmer fluid will give up heat must equal the rate at which the cooler fluid will absorb heat. If the fluid used in the calculation does not change phase, the rate of heat transfer  $q$  can be calculated from the rate of flow, the specific heat, and the entering and leaving temperatures of the fluid. If the fluid evaporates or condenses the rate of heat transfer  $q$  can be calculated from the rate of flow and the heat of vaporization of the fluid. However, any heat given up or absorbed by the fluid as the result of its being superheated or subcooled must also be included.

It may be noted that the temperatures and the rates of flow of the two fluids may not all be chosen arbitrarily, since (a) the values must be such that the heat balance is satisfied, and (b) at every section of the equipment the temperature of the warmer fluid must be higher than the temperature of the cooler fluid. For example, if neither fluid changes phase and if the entering and leaving temperatures of the two fluids are specified the rate of flow of either of the fluids may be assigned any value desired, but the rate of flow of the other fluid must then be determined from the heat balance.

2 The over-all coefficient  $U$  can be calculated from the film coefficients for the two fluids and these can be determined from the charts and tables given in Chap IV. The method of using these charts and tables is explained in Sec 32.

The film coefficients for all types of convection depend upon the temperature of the fluid. If the fluid does not change phase, the temperature of the fluid changes as the fluid flows through the equipment, and consequently the film coefficient also changes. A sufficiently accurate average value can usually be obtained, however, if the film coefficient is evaluated at the *average temperature* of the fluid, the average temperature being taken as the arithmetic average of the temperatures at which the fluid enters and leaves the equipment.

If the heat-transfer surface is to consist of tubes, it may be necessary to select tentative values for the size and the number of the tubes before the film coefficients can be determined. For example, the film coefficient for the fluid flowing through the tubes usually depends upon the velocity of the fluid, which cannot be calculated until the size and the number of the tubes have been selected. If these values are selected in advance, only the length of the tubes remains to be calculated after the amount of heat-transfer surface required has been determined by Eq. (24). If the tube length turns out to be either too long or too short for practical purposes, a new value for either the size or the number of the tubes must be selected, and the calculations repeated.

Having determined the film coefficients for the two fluids, the over-all coefficient  $U$  can be calculated by the equations given in Sec. 25, or it can be evaluated approximately by Fig. 15. Allowance for scale can be made by Eq. (29).

3. The *mean* temperature difference between the two fluids must be used for  $\Delta t$ , and this can be calculated as explained in Sec. 27.

Problems illustrating these calculations are given in the next chapter.



## CHAPTER IV

### CONVECTION COEFFICIENTS

**29 Introduction**—Charts and tables for determining film coefficients for various kinds of convection are given in this chapter. Charts and tables for determining over-all coefficients are not given, since these coefficients can be readily calculated from the film coefficients by means of the equations given in Sec. 25 of Chap. III.

Experimentally, film coefficients are more difficult to measure than over-all coefficients, because the temperature of the retaining wall must be measured in addition to the temperatures of the fluids. However, this increased experimental difficulty is compensated for by the fact that since film coefficients depend upon fewer variables, the results are more easily correlated. Regarding the correlation of experimental data, the problem of correlation can frequently be simplified by means of dimensional analysis. Dimensional analysis may not only indicate how the number of variables can be reduced but may also permit the results obtained with one or two fluids to be used to predict values for other fluids. The simplification sometimes made possible is illustrated by the example given in the next section.

**30 An Illustration of the Use of Dimensional Analysis**—Consider a sphere submerged in a liquid and rolling down an incline as shown in Fig. 20. The sphere accelerates until the resistance offered by the liquid exactly balances the force  $F$  acting on it in the direction of its motion. Let  $V$  denote the maximum velocity attained by the sphere.

For any given diameter sphere, kind of liquid, and liquid temperature the maximum velocity  $V$  corresponding to different values of the force  $F$  could be determined experimentally by changing the angle of inclination  $\alpha$ . If the resulting values of  $V$  were plotted against  $F$ , a separate curve would be obtained for each combination of diameter, kind of liquid, and liquid temperature used. On the other hand, if the results were plotted on the coordinates suggested by dimensional analysis, all the data would fall upon a single curve.

The maximum velocity  $V$  evidently depends upon the force  $F$ , the diameter of the sphere  $D$ , and the density  $\rho$  and viscosity  $\mu$  of the liquid, i. e.

$$V = f(F, D, \rho, \mu)$$

This unknown function can be expressed as an exponential series,

$$V = a_1 F^w D^x \rho^y \mu^z + a_2 F^w D^x \rho^y \mu^z + \dots \quad (31)$$

in which each term is of the form  $aF^w D^x \rho^y \mu^z$ , and  $a$ ,  $w$ ,  $x$ ,  $y$ , and  $z$  are dimensionless constants. Each of the terms in the right-hand member of this equation must have the same net dimensions as those of the left-hand member.

In terms of the three fundamental dimensions mass  $m$ , length  $l$ , and time  $\theta$ , the dimensions of the velocity  $V$  are  $l/\theta$ , of the force  $F$  are  $ml/\theta^2$  (mass  $\times$  acceleration), of the diameter  $D$  are  $l$ , of the density  $\rho$  are  $m/l^3$ , and

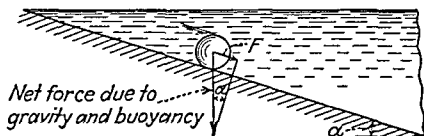


FIG. 20.—Sphere submerged in a liquid and rolling down an incline.

of the viscosity  $\mu$  are  $m/l\theta$  (force  $\times$  distance/area  $\times$  velocity). Hence, equating the dimensions of the general term  $aF^w D^x \rho^y \mu^z$  to the dimensions of  $V$ ,

$$\left(\frac{ml}{\theta^2}\right)^w (l)^x \left(\frac{m}{l^3}\right)^y \left(\frac{m}{l\theta}\right)^z = \frac{l}{\theta},$$

or

$$m^{w+y+z} l^{w+x-3y-z-1} \theta^{-2w-z+1} = 1.$$

This equation will be satisfied if each of the exponents is equal to zero:

$$\begin{aligned} w + y + z &= 0, \\ w + x - 3y - z - 1 &= 0, \\ -2w - z + 1 &= 0, \end{aligned}$$

Solving these three simultaneous equations for  $x$ ,  $y$ , and  $z$  in terms of  $w$ ,

$$\begin{aligned} x &= -1, \\ y &= w - 1, \\ z &= -2w + 1. \end{aligned}$$

Hence, each term of the series must be of the form

$$aF^w D^x \rho^y \mu^z = aF^w D^{-1} \rho^{w-1} \mu^{-2w+1} = a \left(\frac{\mu}{D\rho}\right) \left(\frac{F\rho}{\mu^2}\right)^w$$

and Eq. (31) can be written

$$V = a_1 \left(\frac{\mu}{D\rho}\right) \left(\frac{F\rho}{\mu^2}\right)^{w_1} + a_2 \left(\frac{\mu}{D\rho}\right) \left(\frac{F\rho}{\mu^2}\right)^{w_2} + \dots$$

or

$$\frac{DV\rho}{\mu} = a_1 \left(\frac{F\rho}{\mu^2}\right)^{w_1} + a_2 \left(\frac{F\rho}{\mu^2}\right)^{w_2} + \dots$$

or

$$\frac{DV\rho}{\mu} = f \left(\frac{F\rho}{\mu^2}\right). \quad (32)$$

Thus, dimensional analysis indicates that the quantity  $DV\rho/\mu$  depends only upon the quantity  $F\rho/\mu^2$ , therefore, if these two quantities are used as the coordinates, all the experimental data will fall upon a single curve

Such a curve, determined by experiments with perhaps two or three different diameter spheres and two or three liquids, could be used to predict the maximum velocity  $V$  that a sphere of any other diameter submerged in any other liquid would attain if it were acted upon by any given force  $F$ . The quantity  $F\rho/\mu^2$  would first have to be calculated from the given value of  $F$  and the values of  $\rho$  and  $\mu$  for the liquid. The value of the quantity  $DV\rho/\mu$  corresponding to this value of  $F\rho/\mu^2$  could then be determined from the experimental curve, and, finally, the value of  $V$  could be calculated from this value of  $DV\rho/\mu$ .

TABLE 6—NOMENCLATURE

$c_p$	Specific heat at constant pressure, B t u / (lb) (°F)
$D$	Inside or outside tube diameter, ft
$d$	Inside diameter of pipe, in
$d_c$	Diameter of coil, in
$G$	Weight velocity, lb. / (hr) (sq ft of cross section)
$g$	Acceleration of gravity, $4.18 \times 10^8$ ft per hr <sup>2</sup>
$h$	Film coefficient of convection, B t u / (sq ft) (hr) (°F)
$h_s$	Scale coefficient, B t u / (sq ft) (hr) (°F)
$k$	Thermal conductivity, B t u / (ft) (hr) (°F)
$N$	Length of tube or height of surface, ft
$n$	Number of tubes directly over each other
$P$	Pressure, lb per sq ft abs
$p$	Pressure, atm abs
$R$	Gas constant, ft per °F
$r$	Latent heat of evaporation or condensation, B t u per lb
$T$	Temperature, °F abs
$t$	Temperature, °F.
$t_f$	Film temperature, °F
$\Delta t$	Temperature difference, °F
$V$	Linear velocity, ft per hr
$v$	Specific volume, cu ft per lb
$W$	Rate of condensation, lb / (sq ft) (hr)
$W'$	Rate of flow or rate of condensation, lb per hr per tube
$\beta$	Thermal coefficient of expansion, $1/^\circ\text{F}$
$\mu$	Absolute viscosity, lb / (ft) (hr)
$\nu$	Kinematic viscosity, sq ft per hr.
$\rho$	Density, lb per cu ft
$\sigma$	Surface tension, lb. per ft

**31. Calculation of the Charts and Tables**—The method of dimensional analysis illustrated in the preceding section can also be applied to the phenomenon of convection and has proved of great value in correlating the experimental data. Many investigators have expressed their results in the form of dimen-

sionless equations suggested by it, and certain of these equations have been used to calculate the charts and tables presented in this chapter. These equations together with the other data upon which the charts and tables are based are given in this section. The symbols used are defined in Table 6.

CASE 1. LIQUIDS HEATED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 22 and Tables 8 and 9 are based on the following equation recommended by McAdams:<sup>1</sup>

$$\frac{hD}{k} = 0.0225 \left( \frac{DV\rho}{\mu} \right)^{0.8} \left( \frac{c_p\mu}{k} \right)^{0.4} \quad (33)$$

Each of the physical properties is evaluated at the average temperature of the liquid. This equation is valid for values of  $DV\rho/\mu$  as low as 2,100, provided the value of  $c_p\mu/k$  is less than 10. Otherwise, it is valid only for values of  $DV\rho/\mu$  greater than 10,000. For the range of velocities given in Fig. 22, the liquids affected by these restrictions are listed in Table 10.

Although Eq. (33) is recommended only for horizontal tubes, the data of Stender<sup>2</sup> indicate that it can also be used to calculate conservative values for vertical tubes.

CASE 2. LIQUIDS COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 23 and Tables 11, 12, and 13 are based on an equation exactly like Eq. (33) except with the exponent of the  $c_p\mu/k$  term changed from 0.4 to 0.3. This equation was suggested by the work of Dittus and Boelter.<sup>3</sup>

CASE 3. GASES HEATED OR COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 24 and Tables 14 and 15 are based on Eq. (33). This equation is valid for gases being either heated or cooled, provided the value of  $G = V\rho$  is greater than  $1,650p^{0.645}$ . Values of

$$G_{\min} = (V\rho)_{\min} = 1,650p^{0.645}$$

are given in Table 16.

CASE 4. GASES AT ATMOSPHERIC PRESSURE HEATED OR COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 25 and Tables 17 and 18 are also based on Eq.

<sup>1</sup> McAdams, W. H., "Heat Transmission," p. 169, McGraw-Hill Book Company, Inc., New York, 1933.

<sup>2</sup> *Ind. Eng. Chem.*, vol. 24, p. 1318, 1932.

<sup>3</sup> *Univ. Calif. Pub. Eng.*, vol. 2, p. 443, 1930.

(33), the equation being valid for gases if  $V\rho$  is greater than  $1,650p^{0.44}$ . The gases are assumed to obey the perfect gas law  $Pv = RT$ .

CASE 5 LIQUIDS HEATED OR COOLED OUTSIDE SINGLE TUBES, DIRECTION OF FLOW NORMAL TO TUBE—Figure 26 and Tables 20 and 21 are based on the following equation

$$\frac{hD}{k} = 0.385 \left( \frac{c_p \mu}{k} \right)^{0.3} \left( \frac{DV\rho}{\mu} \right)^{0.56}, \quad (34)$$

each of the physical properties being evaluated at the film temperature  $t_f$ . The form of this equation was suggested by Ulbramer,<sup>1</sup> and the constant 0.385 was determined from Reihers's<sup>2</sup> equation for air

$$\frac{hD}{k} = 0.35 \left( \frac{DV\rho}{\mu} \right)^{0.56}, \quad (35)$$

by taking  $c_p \mu / k$  for air equal to 0.73. Equation (34) is recommended only for values of  $DV\rho/\mu$  greater than 100. No data for liquids are available in this range but the data of Davis<sup>3</sup> on water and oil for the range  $DV\rho/\mu$  from 0.1 to 100 agree very well with the data on air for the same range. For the range of values of velocity, temperature, and diameter given in Fig. 26 and Tables 20 and 21,  $DV\rho/\mu$  always exceeds 100.

CASE 6 GASES HEATED OR COOLED OUTSIDE SINGLE TUBES, DIRECTION OF FLOW NORMAL TO TUBE—Figure 27 and Tables 22 and 23 are based on Eq. (34). This equation is not recommended for values of  $DV\rho/\mu$  less than 100, but for the range of values of velocity, temperature, and diameter given on the chart and in the tables,  $DV\rho/\mu$  always exceeds this minimum value.

CASE 7 GASES AT ATMOSPHERIC PRESSURE HEATED OR COOLED OUTSIDE SINGLE TUBES, DIRECTION OF FLOW NORMAL TO TUBE—Figure 28 and Tables 24 and 25 are based on Eq. (34) and on the assumption that the gases obey the perfect gas law  $Pv = RT$ . For the range of values of velocity, temperature, and diameter given on the chart and in the tables the product  $DV\rho/\mu$  always exceeds the minimum permissible value of 100.

CASE 8 LIQUIDS OR GASES HEATED OR COOLED OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW NORMAL TO TUBES—The

<sup>1</sup> *Forsch. Gebiete Ing.* vol. 3 p. 94, 1932.

<sup>2</sup> *Mitt. Forschungsarb.* vol. 269 p. 20, 1925.

<sup>3</sup> *Phil. Mag.* vol. 47 pp. 972 and 1057, 1924.

data of Pierson, Huge, and Grimson<sup>1</sup> on air indicate that approximate values of the film coefficient  $h$  for fluids flowing normal to banks of tubes can be determined by multiplying the value of  $h$  for flow normal to single tubes by 1.3 if the tubes are staggered and by 1.2 if the tubes are in line.

The data of Reiher on air flowing normal to five rows of staggered tubes also show an increase in the film coefficient for tube bundles compared to the value for single tubes. Although his data for flow normal to five rows of tubes in line show a decrease compared to single tubes, Reiher found that the average value of the film coefficient for the entire tube bundle increased as the number of rows of tubes was increased. Consequently, had he used more than five rows of tubes, he might also have found the average coefficient for the entire bundle to be greater than for a single tube.

CASE 9. LIQUIDS OR GASES HEATED OR COOLED OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW PARALLEL TO TUBES.—For flow parallel to the outside of tubes, Nusselt<sup>2</sup> recommends that film coefficients be determined from the data for flow inside tubes, using an equivalent inside diameter equal to four times the hydraulic radius. Equation (46) and Figs. (29) and (30) are based on this procedure.

CASE 10. LIQUIDS OR GASES HEATED OR COOLED IN ANNULAR SPACES, TURBULENT FLOW.—Film coefficients for fluids flowing through the annular space of a double-pipe heat exchanger, in which heat transfer is taking place between the fluid in the inner pipe and the fluid in the annular space, can be determined from the data for flow inside tubes, provided an equivalent diameter equal to four times the hydraulic radius is used for the inside diameter. Equation (47) is based on this procedure.

CASE 11. LIQUIDS HEATED OUTSIDE SINGLE HORIZONTAL TUBES, NATURAL CONVECTION.—Figure 31 and Tables 26 and 27 are based on the following equation recommended by McAdams:<sup>3</sup>

$$\frac{hD}{k} = 0.525 \left[ \left( \frac{D^3 \rho^2 \beta g (\Delta t)}{\mu^2} \right) \left( \frac{c_p \mu}{k} \right) \right]^{0.25}, \quad (36)$$

<sup>1</sup> *Trans. A.S.M.E.*, vol. 59, pp. 563, 573, and 583, 1937.

<sup>2</sup> *Z. Ver. deut. Ing.*, vol. 57, p. 199, 1913, and vol. 61, p. 685, 1917.

<sup>3</sup> McADAMS, *op. cit.*, p. 249.

each of the physical properties being evaluated at the film temperature  $t_f$ . This equation is valid only for values of

$$\left[ \left( \frac{D^3 p^2 \beta g (\Delta t)}{\mu^2} \right) \left( \frac{c_p \mu}{k} \right) \right]$$

greater than 1,000. This value is exceeded for all the liquids listed for the ranges of temperature difference, film temperature, and diameter given on the chart and in the tables.

**CASE 12 GASES HEATED OUTSIDE SINGLE HORIZONTAL TUBES, NATURAL CONVECTION**—Figure 32 and Tables 28, 29, and 30 are based on Eq. (36) and on the assumption that the gases obey the perfect gas law  $Pv = RT$ . For the ranges of temperature difference, film temperature, diameter, and pressure given, none of the gases listed falls below the lower limit for which Eq. (36) is valid.

**CASE 13 LIQUIDS HEATED INSIDE OR OUTSIDE VERTICAL TUBES OR ON VERTICAL PLATES. LOW VELOCITIES OR NATURAL CONVECTION ONLY**—Figure 33 and Table 31 are based on the following equation of Colburn and Hougen:<sup>1</sup>

$$h = 0.128 \left( \frac{k^2 p^2 c_p \beta g (\Delta t)}{\mu} \right)^{1/4} \quad (37)$$

Each of the physical properties is evaluated at the film temperature  $t_f$ . This equation may be used either for natural convection or for forced convection, provided the velocities are low. The equation is based on data for *upward* flow inside tubes, but it may also be used to predict conservative values for *downward* flow. It may also be used for predicting film coefficients for fluids outside tubes, since it agrees very well with the equation of Rice<sup>2</sup> for air heated by natural convection outside pipes.

**CASE 14 GASES HEATED INSIDE OR OUTSIDE VERTICAL TUBES OR ON VERTICAL PLATES, NATURAL CONVECTION**—Figure 34 and Tables 32 and 33 are also based on Eq. (37) and on the assumption that the gases obey the perfect-gas law  $Pv = RT$ .

**CASE 15 LIQUIDS HEATED OR COOLED INSIDE COILS, TURBULENT FLOW**—For any given velocity, the flow inside coiled tubes is considerably more turbulent than that inside straight tubes. This results in an increase in the value of the film coefficient  $h$ ,

<sup>1</sup> *Ind. Eng. Chem.*, vol. 22, p. 522, 1930.

<sup>2</sup> *Trans. Am. Inst. Elec. Eng.*, vol. 43, p. 131, 1924.

the exact amount depending upon the radius of the coil. Richter<sup>1</sup> found that over-all coefficients for heat transfer from water to water inside a helical double-pipe exchanger were about 20 per cent larger than over-all coefficients for straight double-pipe exchangers. This suggests that approximate values for the film coefficient  $h$  can be obtained by multiplying the value of  $h$  for straight tubes by a factor such as 1.2.

CASE 16. GASES HEATED OR COOLED INSIDE COILS, TURBULENT FLOW.—Based on the work of Jeschke, McAdams<sup>2</sup> suggests that values of the film coefficient  $h$  for the turbulent flow of gases inside coils be obtained by multiplying the value of  $h$  for straight tubes by the factor

$$\left[ 1 + 3.54 \left( \frac{d}{d_c} \right) \right],$$

where  $d/d_c$  is the ratio of the diameters of the pipe and the coil.

CASE 17. LIQUIDS OR GASES HEATED OR COOLED OUTSIDE COILS, NATURAL OR FORCED CONVECTION.—The data of Gibson<sup>3</sup> on air flowing normal to coils agree very well with Eq. (35). This suggests that film coefficients for fluids flowing outside coils may be assumed approximately equal to the film coefficients for fluids flowing normal to single tubes.

CASE 18. AIR HEATED ON HORIZONTAL PLATES, NATURAL CONVECTION.—Heating air by natural convection on large flat plates, Griffiths and Davis<sup>4</sup> found that the coefficient  $h$  for horizontal plates facing *upward* was about 27 per cent higher than for vertical plates; and for horizontal plates facing *downward*, about 33 per cent lower.

CASE 19. PETROLEUM OILS HEATED INSIDE HORIZONTAL OR VERTICAL TUBES, STREAMLINE FLOW.—Figure 35 and Tables 34 and 35 are based on the following empirical equation suggested by McAdams:<sup>5</sup>

$$\frac{hD}{k} = 6.2 \left( \frac{W'c_p}{kN} \right)^{0.2}, \quad (38)$$

<sup>1</sup> *Am. Inst. Chem. Eng.*, vol. 12, Part II, p. 147, 1919.

<sup>2</sup> McADAMS, *op. cit.*, p. 179.

<sup>3</sup> *Phil. Mag.*, Ser. 6, vol. 47, p. 324, 1924.

<sup>4</sup> Food Investigation Board, *Dept. Sci. Ind. Research, Special Rept. 9*, His Majesty's Stationery Office, London, 1922.

<sup>5</sup> McADAMS, *op. cit.*, p. 210.



in which the coefficient  $h$  is based on the arithmetic mean of the terminal temperature differences, and  $c_p$  and  $k$  are evaluated at the average temperature of the oil after mixing. This equation is valid only if  $W'c_p/k\lambda$  exceeds 30 and if  $DV\rho/\mu$  is less than 2,100. For the range of gravities and  $W'/\lambda$  given in Fig. 35, the product  $W'c_p/k\lambda$  always exceeds 30. The values of  $W'/D$  for which  $DV\rho/\mu$  equals 2,100 are given in Table 36.

**CASE 20. PETROLEUM OILS COOLED INSIDE HORIZONTAL OR VERTICAL TUBES STREAMING FLOW**—The data plotted by McAdams<sup>1</sup> on cooling hydrocarbon oils flowing at low velocities inside either horizontal or vertical tubes show that approximate values of the film coefficient  $h$  for cooling can be obtained by multiplying the values of  $h$  for heating calculated by Eq. (38), by 0.7.

**CASE 21. CONDENSATION OF PURE SATURATED VAPORS ON HORIZONTAL TUBES**—Figure 36 and Tables 37, 38, and 39 are based on theoretical equation of Nusselt<sup>2</sup> for filmwise condensation of pure saturated vapors on horizontal tubes

$$h = 0.725 \left[ \frac{r\rho^*k^3g}{n\mu D(\Delta t)} \right]^{1/4} \quad (39)$$

and on the equation

$$h(\Delta t) = W'r \quad (40)$$

The latent heat of condensation  $r$  is evaluated at the temperature of the vapor but the other physical properties are evaluated at the temperature of the condensate film. Equation (39) gives conservative values; the deviation from test data increasing with the rate of condensation.

**CASE 22. CONDENSATION OF PURE SATURATED VAPORS ON VERTICAL TUBES OR PLATES**—Figure 37 and Tables 41 and 42 are based on the following equation given by Kirkbride<sup>3</sup> for filmwise condensation of pure vapors on vertical tubes

$$h \left( \frac{\mu^2}{k^3\rho^2g} \right)^{1/4} = 0.0084 \left( \frac{W}{\mu D} \right)^{0.4} \quad (41)$$

Each of the physical properties is evaluated at the temperature of the condensate film. Figure 36 and Tables 37 and 43 are

<sup>1</sup> McAdams *op cit* p. 210.

<sup>2</sup> *Z. Ver. dest. Ing.* vol. 60 pp. 541 and 569, 1916.

<sup>3</sup> *Trans. Am. Inst. Chem. Eng.* vol. 32 p. 170, 1936.

based on the theoretical equation of Nusselt for filmwise condensation of pure saturated vapors on vertical surfaces

$$h = 0.943 \left[ \frac{r \rho^2 k^3 g}{N \mu (\Delta t)} \right]^{1/4} \quad (42)$$

and on Eq. (40). The latent heat of condensation is evaluated at the temperature of the vapor, and the other physical properties are evaluated at the temperature of the condensate film. Equations (41) and (42) both yield the same value of the film coefficient  $h$  when  $W'/\mu D$  is equal to 1,020. Equation (41) is recommended if  $W'/\mu D$  is greater than 1,020, and Eq. (42) is recommended if  $W'/\mu D$  is smaller. The values of  $WN$  for which  $W'/\mu D$  is equal to 1,020 are given in Table 40.

CASE 23. LIQUIDS BOILING ON HORIZONTAL OR VERTICAL PLATES.—Figure 38 and Table 44 are based on the equation of Jakob<sup>1</sup> for evaporation on flat plates free from scale, occurring after the equipment has been in operation for some time:

$$\frac{h}{k} \left( \frac{\sigma}{\rho' - \rho''} \right)^{0.5} = 31.6 \frac{\nu_s}{\nu} \left[ \left( \frac{\sigma}{\sigma_a} \right) \left( \frac{\rho_a' - \rho_a''}{\rho' - \rho''} \right) \frac{h(\Delta t)}{\rho_a'' r_a w_a} \right]^{0.8}, \quad (43)$$

where the prime denotes saturated liquid; the double prime, saturated vapor; the subscript  $a$ , atmospheric pressure; the subscript  $s$ , saturated water at 212°F.; and  $w_a$  is equal to 918 ft. per hr. and represents the product of the mean frequency of formation of bubbles at any point on the heating surface times the mean diameter of the vapor bubbles leaving the surface. This equation gives quite conservative values of the film coefficient.

CASE 24. LIQUIDS BOILING INSIDE TUBES.—According to Jakob, values of the coefficient  $h$  for evaporation in tubes are about 25 per cent higher than for evaporation on flat plates.

CASE 25. SCALE COEFFICIENTS.—McAdams<sup>2</sup> suggests that the scale coefficients given be used if specific information is not available.

32. Use of the Charts and Tables.—Values of the film coefficient  $h$  for various fluids under a number of different conditions can be readily determined by means of the charts and tables that follow. Table 7 will be found useful as an index to the various cases covered.

<sup>1</sup> *Tech. Bull. Armour Inst. Tech.*, vol. 2, no. 1, 1939.

<sup>2</sup> McADAMS, *op. cit.*, p. 151.

TABLE 7—OUTLINE OF THE KINDS OF CONVECTION FOR WHICH FILM COEFFICIENTS ARE GIVEN

a Convection involving no change of phase

Kind of fluid	Kind of surface	Position of surface horizontal or vertical	Kind of fluid motion	Fluid heated or cooled	Case No	Page
Liquids	Inside tubes	Horizontal	Turbulent	Heated Cooled	1 2	80 82
			Streamline (pet oils only)	Heated Cooled	19 20	105 105
		Vertical	Turbulent	Heated Cooled	1 2	80 82
			Streamline (pet oils only)	Heated Cooled	19 20	105 105
			Natural conv	Heated	13	100
	Outside single tubes	Horizontal	Normal to tube Natural conv	Either Heated	5 11	88 96
		Vertical	Normal to tube Natural conv	Either Heated	5 13	88 100
	Outside tube bundles	Horizontal	Normal to tube Parallel to tubes	Either Either	8 9	94 94
		Vertical	Normal to tubes Parallel to tubes Natural conv	Either Either Heated	8 9 13	94 94 100
	Annular spaces	Either	Turbulent	Either	10	94
	Inside coils	Either	Turbulent	Either	15	104
	Outside coils	Either	Forced conv Natural conv	Either Heated	17 17	104 104
	Flat plates	Vertical	Natural conv	Heated	13	100
Gases	Inside tubes	Horizontal	Turbulent	Either	3 or 4	84
		Vertical	Turbulent Natural conv	Either Heated	3 or 4 14	84 102
	Outside single tubes	Horizontal	Normal to tube Natural conv	Either Heated	6 or 7 12	90 98
		Vertical	Normal to tube Natural conv	Either Heated	6 or 7 14	90 102
	Outside tube bundles	Horizontal	Normal to tubes Parallel to tubes	Either Either	8 9	94 94
		Vertical	Normal to tubes Parallel to tubes Natural conv	Either Either Heated	8 9 14	94 94 102
	Annular spaces	Either	Turbulent	Either	10	94
	Inside coils	Either	Turbulent	Either	16	104
	Outside coils	Either	Forced conv Natural conv	Either Heated	17 17	104 104
	Flat plates	Horizontal (air only)	Natural conv	Heated	18	104
		Vertical	Natural conv	Heated	14	102

## b Condensation of saturated vapors

Kind of surface	Position of surface, horizontal or vertical	Case No	Page
Inside tubes	Vertical	22	108
Outside tubes	{ Horizontal	21	106
	Vertical	22	108
Flat plates	Vertical	22	108

## c Evaporation of liquids

Kind of surface	Position of surface, horizontal or vertical	Case No	Page
Inside tubes	Vertical	24	110
Flat plates	Either	23	110

## d Scale coefficients, Case 25, page 112

The same method of presentation is used throughout: Values of the film coefficient  $h_0$  at some arbitrarily chosen "base" condition are given on the charts, and correction factors for changing these values to conditions other than the base conditions are given in the tables accompanying the charts. For example, in Case 1 the film coefficient  $h$  for any of the liquids listed, flowing at any velocity  $V$  and temperature  $t$  inside a tube of any inside diameter  $d$ , can be found from the equation

$$h = h_0 \times F_t \times F_d, \quad (44)$$

where  $h$  = the desired film coefficient, B.t.u./(sq. ft.)(hr.)(°F.).

$h_0$  = the film coefficient for the given liquid at the "base" conditions and at the velocity  $V$ , read from Fig. 22.

$F_t$  = the temperature correction factor corresponding to the given temperature  $t$ , read from Table 8.

$F_d$  = the diameter correction factor corresponding to the given inside tube diameter  $d$ , read from Table 9.

Many of the charts and tables involve either the temperature difference  $\Delta t$  between the surface of the retaining wall and the main body of the fluid or the *film temperature*  $t_f$ , which is arbitrarily defined as the arithmetic average of the temperature of the surface of the retaining wall and the temperature of the main body of the fluid. In order to evaluate either of these quantities, the temperature of the surface of the retaining wall must be known. Sufficiently accurate results can frequently be obtained by assuming some value that appears reasonable for this temperature. If more accurate results are desired, the

exact temperature of the retaining wall can be calculated by trial and error as follows. Neglecting the temperature drop through the retaining wall itself and using the nomenclature shown on Fig 21, (1) a tentative value is assumed for the temperature  $t_w$  of the retaining wall. (2) Based on this value, the values of  $\Delta t_1$ ,  $\Delta t_2$  and  $t_f$  are calculated. (3) The film coefficients  $h_1$  and  $h_2$  corresponding to these values are determined from the

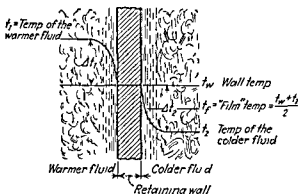


FIG. 21—Temperature gradient through two fluids separated by a retaining wall

charts and tables. (4) Using these values of  $h_1$  and  $h_2$ , a more accurate value of  $t_w$  is calculated by the equation

$$t_w = \frac{h_1 t_1 + h_2 t_2}{h_1 + h_2} \quad (45)$$

This value of  $t_w$  will usually be sufficiently accurate for practical purposes. Using it values of  $\Delta t_1$ ,  $\Delta t_2$  and  $t_f$  can again be calculated, and the final value of  $h_1$  and  $h_2$  can then be determined.

Equation (45) is based on the simplifying assumption that the temperature drop through the retaining wall is negligible and that the areas of the two surfaces of the wall are approximately equal. By Eq. (23) in Chap. III,

$$q = h_1 A (t_1 - t_w) = h_2 A (t_w - t_2)$$

Equation (45) is obtained by solving the last equation for  $t_w$ .

**Illustrative Problem 1 (Shell and tube Heat Exchanger)**—Design a shell and tube heat exchanger to cool 40 gal of water per minute from 55°F by means of 60 gal per min of calcium chloride brine (25 per cent solution) initially at 30°F. The tubes are to be  $\frac{3}{4}$  in. o.d. with No. 16 B.W.G. walls and are to be made of brass.

*Data.*—Specific heat of water = 1.00 B.t.u./(lb.)(°F.); specific heat of the brine = 0.68 B.t.u./(lb.)(°F.); density of water = 8.35 lb. per gal.; density of the brine = 10.3 lb. per gal. Conversion factor: 1 cu. ft. = 7.48 gal.

*Solution.*—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature drop of the water. Thus,

$$\begin{aligned} q &= (40 \times 8.35 \times 60) \times 1.00 \times (55 - 45) \\ &= 200,000 \text{ B.t.u. per hr.} \end{aligned}$$

Since this is the rate at which the brine will absorb heat, the final temperature of the brine can be calculated by the equation

$$200,000 = (60 \times 10.3 \times 60) \times 0.68 \times (t - 30),$$

or

$$t = 38^\circ\text{F.}$$

Let the brine flow through the tubes and the water flow through the shell. The velocity of the brine should be at least 3 ft. per sec. in order to assure turbulent flow (see Table 10). This velocity will be obtained if 19 tubes are used per pass. Thus, since the internal cross-sectional area of each tube is 0.302 sq. in.,

$$\frac{60}{7.48 \times 60} = \frac{19 \times 0.302}{144} \times V,$$

or

$$V = 3.4 \text{ ft. per sec.}$$

*First Design.*—Let each liquid make a single pass through the exchanger. If the tubes are spaced at the apexes of 1-in. equilateral triangles, the shell may be made of 5-in. pipe. Since the internal cross-sectional area of 5-in. pipe is 20.0 sq. in. and the external cross-sectional area of each tube is 0.442 sq. in., the velocity of the water can be calculated by the equation

$$\frac{40}{7.48 \times 60} = \frac{(20.0 - 19 \times 0.442)}{144} \times V,$$

or

$$V = 1.1 \text{ ft. per sec.}$$

The film coefficient  $h_1$  for the brine can be determined by Case 1. Thus, from Fig. 22,  $h_0 = 680$  B.t.u./(sq. ft.)(hr.)(°F.); from Table 8,  $F_t = 0.68$  (average temperature of the brine =  $34^\circ\text{F.}$ ); and from Table 9,  $F_d = 1.10$  (inside tube diameter = 0.620 in.). Hence,

$$\begin{aligned} h_1 &= 680 \times 0.68 \times 1.10 \\ &= 510 \text{ B.t.u./(sq. ft.)(hr.)(°F.).} \end{aligned}$$

The film coefficient  $h_2$  for the water can be determined by Cases 9 and 2. Thus, referring to Fig. 29,  $x = 1$  in.,  $d = 0.75$  in.,  $x/d = 1.33$ , and therefore  $d_s = 0.72$  in. Then, from Fig. 23,  $h_0 = 290$  B.t.u./(sq. ft.)(hr.)(°F.); from Table 11,  $F_t = 0.66$  (average temperature of the water =  $50^\circ\text{F.}$ ); and from Table 12,  $F_d = 1.08$  (equivalent inside diameter = 0.72 in.). Hence,

$$\begin{aligned} h_2 &= 290 \times 0.66 \times 1.08 \\ &= 207 \text{ B.t.u./(sq. ft.)(hr.)(°F.).} \end{aligned}$$

The mean temperature difference between the steam and the water can be determined from Fig. 17 [or calculated by Eq. (30)]. Thus,

$$\Delta t = 227 - 60 = 167^{\circ}\text{F},$$

$\Delta t' = 227 - 180 = 47^{\circ}\text{F}$ , and  $\Delta t_m = 95^{\circ}\text{F}$ . The mean temperature of the water may be taken, therefore, as  $227 - 95 = 132^{\circ}\text{F}$ .

In order to limit the pressure drop, the velocity of the water in the tubes should not exceed 2.5 ft per sec. This requirement is satisfied if 28 tubes are used per pass. Thus, since the cross-sectional area of each tube is 0.334 sq in.,

$$\frac{4000}{7.48 \times 3.600} = \frac{28 \times 0.334}{144} \times V,$$

or

$$V = 2.3 \text{ ft per sec}$$

In order to limit the length of the tubes, let the heater be two pass.

The film coefficient  $h_1$  for the water can be determined by Case 1. From Fig. 22,  $h_0 = 610 \text{ Btu}/(\text{sq ft})(\text{hr})(^{\circ}\text{F})$ , from Table 8

$$F_1 = 1.14 \text{ (at } 132^{\circ}\text{F)},$$

and from Table 9  $F_2 = 1.09$  (inside tube diameter = 0.652 in.). Hence,

$$\begin{aligned} h_1 &= 610 \times 1.14 \times 1.09 \\ &= 760 \text{ Btu}/(\text{sq ft})(\text{hr})(^{\circ}\text{F}) \end{aligned}$$

The film coefficient  $h_2$  for the steam can be determined by Case 21. Since this coefficient depends upon the tube-wall temperature  $t_w$ , a trial-and-error solution is necessary.

As a first trial assume  $t_w = 200^{\circ}\text{F}$ . The inside tube surface  $A_1$  corresponding to this value of  $t_w$  can be calculated from the film coefficient  $h_1$  by Eq. (23)

$$4000/0.000 = 760 \times A_1 \times (200 - 132),$$

or

$$A_1 = 77.5 \text{ sq ft}$$

The corresponding outside tube surface is

$$77.5 \times \left( \frac{0.750}{0.652} \right) = 89.2 \text{ sq ft}$$

and the rate of condensation of the steam

$$\Pi = \frac{4170}{89.2} = 47 \text{ lb}/(\text{sq ft})(\text{hr})$$

Then from Fig. 36  $h_0 = 1450 \text{ Btu}/(\text{sq ft})(\text{hr})(^{\circ}\text{F})$ , from Table 37  $F_1 = 1.47$  (at  $t_w = 200^{\circ}\text{F}$ ), from Table 38  $F_2 = 1.10$ , and from Table 39  $F_3 = 0.55$  (assuming  $n = 6$ ). Hence,

$$\begin{aligned} h_2 &= 1450 \times 1.47 \times 1.10 \times 0.55 \\ &= 1290 \text{ Btu}/(\text{sq ft})(\text{hr})(^{\circ}\text{F}) \end{aligned}$$

The tube wall temperature can now be calculated by Eq. (45):

$$t_w = \frac{760 \times 132 + 1290 \times 227}{760 + 1290} \\ = 192^\circ\text{F.}$$

Since this does not agree with the value initially assumed, the calculations must be repeated.

As a second trial, assume  $t_w = 192^\circ\text{F.}$  By Eq. (23),

$$4,000,000 = 760 \times A_1 \times (192 - 132),$$

or

$$A_1 = 87.7 \text{ sq. ft.}$$

The corresponding outside surface  $A_2 = 87.7 \times (0.750/0.652) = 101 \text{ sq. ft.,}$  and the rate of condensation

$$W = \frac{4,170}{101} = 41 \text{ lb./ (sq. ft.) (hr.).}$$

From Fig. 36,  $h_0 = 1500 \text{ B.t.u./ (sq. ft.) (hr.) (}^\circ\text{F.)}$ ; and from Table 37,

$$F_t = 1.43 \text{ (at } t_w = 192^\circ\text{F.)}$$

Hence,

$$h_2 = 1500 \times 1.43 \times 1.10 \times 0.55 \\ = 1300 \text{ B.t.u./ (sq. ft.) (hr.) (}^\circ\text{F.)}$$

and by Eq. (45),

$$t_w = \frac{760 \times 132 + 1300 \times 227}{760 + 1300} \\ = 192^\circ\text{F.}$$

Since this agrees with the value initially assumed, the preceding calculations are correct, and the outside tube surface required is 101 sq. ft. Since 5.093 lin. ft. of  $\frac{3}{4}$ -in. tubing have 1 sq. ft. of outside surface, the total length of tubing required is  $5.093 \times 101 = 513 \text{ ft.,}$  and the length of each tube must be  $513/56 = 9.2 \text{ ft.}$  (Note that in the foregoing solution the temperature drop through the tube wall has been neglected.)

**Illustrative Problem 3 (Oil Heater).**—Design a shell-and-tube type heater to heat 800 gal. per hr. of fuel oil having a gravity of  $30^\circ \text{ A.P.I.}$  and a viscosity of 150 Saybolt Universal seconds at  $100^\circ\text{F.}$  from  $50$  to  $200^\circ\text{F.}$  by means of steam at 10 lb. per sq. in. gauge pressure (temperature =  $242^\circ\text{F.}$ ). The tubes are to be  $\frac{3}{4}$  in. o.d. with No. 16 B.W.G. walls, and the oil is to flow through the tubes.

**Data.**—Mean specific heat of the oil =  $0.47 \text{ B.t.u./ (lb.) (}^\circ\text{F.)}$ , and density of the oil =  $7.30 \text{ lb. per gal.}$

**Solution.**—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature rise of the oil. Thus,

$$q = (800 \times 7.30) \times 0.47 \times (200 - 50) \\ = 410,000 \text{ B.t.u. per hr.}$$



The length of the heater will depend upon the number of tubes used. Tentatively, let the heater consist of 90 tubes inclosed in a 12-in. shell, and let the oil make two passes through the tubes.

Since the flow of the oil will be streamline (see Table 36), the film coefficient  $h_1$  for the oil can be determined by Case 19. Since the value of  $h_2$  depends upon the length of the tubes, a trial and-error solution is necessary. As a first trial, assume that the tubes will be 10 ft long. Then, remembering that the oil will make two passes through the tubes,

$$W' = \frac{(800 \times 7.30)}{45} = 130 \text{ lb per hr per tube,}$$

$N = 2 \times 10 = 20 \text{ ft}$ ,  $W'/N = 130/20 = 6.5 \text{ lb/(hr)(tube)(ft)}$ , and from Fig. 35  $h_1 = 12.0 \text{ Btu/(sq ft)(hr)(}^\circ\text{F)}$ . From Table 34,  $F_1 = 1.00$ , and from Table 35,  $F_2 = 1.62$  (inside tube diameter = 0.620 in). Hence,

$$\begin{aligned} h_1 &= 12.0 \times 1.00 \times 1.62 \\ &= 19.5 \text{ Btu/(sq ft)(hr)(}^\circ\text{F)} \end{aligned}$$

The film coefficient  $h_2$  for the condensing steam can be determined by Case 21. However, since it will be quite large compared with  $h_1$ ,  $h_2$  will have very little effect on the over all coefficient  $U$  [i.e., the  $1/h_2$  term in Eq. (27) will be almost negligible], so its exact value need not be determined. Instead, assume  $h_2 = 3000 \text{ Btu/(sq ft)(hr)(}^\circ\text{F)}$ .

The over all coefficient  $U$ , based on the outside tube surface, can be calculated by Eq. (27). Neglecting the  $L/k$  term,

$$\frac{1}{U} = \frac{0.750}{19.5 \times 0.620} + \frac{1}{3000}$$

or

$$U = 16.1 \text{ Btu/(sq ft)(hr)(}^\circ\text{F)}$$

Since the film coefficient  $h_1$  for the oil is based on the *arithmetic* mean of the terminal temperature differences, the mean temperature difference between the oil and the steam may be calculated as follows:

$$\Delta t = 242 - 50 = 192^\circ\text{F},$$

$$\Delta t = 242 - 200 = 42^\circ\text{F} \quad \text{and} \quad \Delta t_{\text{mean}} = (192 + 42)/2 = 117^\circ\text{F}$$

The outside tube surface required can now be calculated by Eq. (24)

$$410,000 = 16.1 \times A \times 117,$$

or

$$A = 218 \text{ sq ft}$$

Since 5.093 lin. ft. of  $\frac{3}{4}$  in. tubing is required per square foot of outside surface, the total length of tubing required is  $5.093 \times 218 = 1,110 \text{ ft}$ , and the length of each tube must be  $1,110/90 = 12.3 \text{ ft}$ . Although this value does not agree with the value assumed in determining  $h_1$ , it is not necessary to repeat the calculations because changing the tube length from 10 to 12.3 ft. would have an almost negligible effect on  $h_1$  (see Fig. 35). Since the tubes are of a reasonable length, this design would be satisfactory.

**Illustrative Problem 4 (Double-pipe Heat Exchanger).**—A double-pipe heat exchanger is to be used to cool 10 gal. of water per minute from 55 to 45°F. by means of 10 gal. per min. of calcium chloride brine (25 per cent solution) initially at 10°F. How long must the exchanger be if it is made of  $1\frac{1}{4}$ - and  $2\frac{1}{2}$ -in. wrought-iron pipe?

*Data.*—Specific heat of water = 1.00 B.t.u./ (lb.) (°F.); specific heat of the brine = 0.68 B.t.u./ (lb.) (°F.); density of water = 8.35 lb. per gal.; density of the brine = 10.3 lb. per gal. Conversion factor:

$$1 \text{ cu. ft.} = 7.48 \text{ gal.}$$

*Solution.*—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature drop of the water:

$$\begin{aligned} q &= (10 \times 8.35 \times 60) \times 1.00 \times (55 - 45) \\ &= 50,000 \text{ B.t.u. per hr.} \end{aligned}$$

Since this is the rate at which the brine will absorb heat, the final temperature of the brine can be calculated by the equation

$$50,000 = (10 \times 10.3 \times 60) \times 0.68 \times (t - 10),$$

or

$$t = 22^\circ\text{F.}$$

Let the brine flow through the inner pipe, and the water through the annular space. The velocity of the brine can be calculated by the equation (internal cross-sectional area of a  $1\frac{1}{4}$ -in. pipe = 1.495 sq. in.)

$$\frac{10}{7.48 \times 60} = \frac{1.495}{144} \times V,$$

or

$$V = 2.2 \text{ ft. per sec.}$$

Similarly, the velocity of the water can be calculated by the equation (internal cross-sectional area of a  $2\frac{1}{2}$ -in. pipe = 4.788 sq. in.; external cross-sectional area of a  $1\frac{1}{4}$ -in. pipe = 2.164 sq. in.)

$$\frac{10}{7.48 \times 60} = \frac{(4.788 - 2.164)}{144} \times V,$$

or

$$V = 1.2 \text{ ft. per sec.}$$

The film coefficient  $h_1$  for the brine can be determined by Case 1. Thus, from Fig. 22,  $h_0 = 470$  B.t.u./ (sq. ft.) (hr.) (°F.); from Table 8,

$$F_i = 0.60 \text{ (average brine temperature} = 16^\circ\text{F.)};$$

and from Table 9,  $F_d = 0.94$  (inside pipe diameter = 1.38 in.). Hence,

$$\begin{aligned} h_1 &= 470 \times 0.60 \times 0.94 \\ &= 265 \text{ B.t.u./ (sq. ft.) (hr.) (°F.).} \end{aligned}$$

The film coefficient  $h_2$  for the water can be determined by Cases 10 and 2. Thus, by Eq. (47) (inside diameter of a  $2\frac{1}{2}$ -in. pipe = 2.469 in.; outside

diameter of a  $1\frac{1}{4}$  in pipe = 1.660 in ),

$$d_s = \frac{(2.469)^2 - (1.660)^2}{1.660} \\ = 2.02 \text{ in}$$

From Fig 23,  $h_0 = 310 \text{ Btu}/(\text{sq ft})(\text{hr})(^\circ\text{F})$ , from Table 11,

$$F_1 = 0.66 \text{ (average water temperature} = 50^\circ\text{F)},$$

and from Table 12,  $F_2 = 0.87$  (for  $d_s = 2.02$  in) Hence,

$$h_1 = 310 \times 0.66 \times 0.87 \\ = 178 \text{ Btu}/(\text{sq ft})(\text{hr})(^\circ\text{F})$$

The over all coefficient  $U$  based on the outside surface of the inner pipe can be calculated by Eq (27) Since the pipe wall is 0.140 in thick, and since the thermal conductivity of wrought iron is about  $35 \text{ Btu}/(\text{ft})(\text{hr})(^\circ\text{F})$ ,

$$\frac{1}{U} = \frac{1.660}{265 \times 1.380} + \frac{0.140}{12 \times 35} + \frac{1}{178}$$

or

$$U = 95 \text{ Btu}/(\text{sq ft})(\text{hr})(^\circ\text{F})$$

For counterflow, the terminal temperature differences are

$$\Delta t' = 45 - 10 = 35^\circ\text{F}$$

and  $\Delta t'' = 55 - 22 = 33^\circ\text{F}$ , so the mean temperature difference =  $34^\circ\text{F}$

The pipe surface required can now be calculated by Eq (24)

$$50,000 = 95 \times A \times 34$$

or

$$A = 15.5 \text{ sq ft}$$

Since 2.301 lin ft of  $1\frac{1}{4}$  in pipe have 1 sq ft of outside surface, the total length of the heat exchanger must be  $2.301 \times 15.5 = 35.6$  ft The exchanger could consist, for example, of four sections each 9 ft long

**Illustrative Problem 5 (Pipe Coil)**—One hundred cubic feet of free air per minute, compressed to a pressure of 50 lb per sq in gauge is to be cooled from  $210$  to  $90^\circ\text{F}$  in a coil submerged in water at  $80^\circ\text{F}$  How long must the coil be if it is made of  $1\frac{1}{4}$  in steel pipe bent into a 2-ft diameter helix?

**Data**—Specific heat of air at constant pressure =  $0.24 \text{ Btu}/(\text{lb})(^\circ\text{F})$ , density of the free air =  $0.075 \text{ lb per cu ft}$

**Solution**—The weight of air flowing through the coil will be

$$100 \times 60 \times 0.075 = 450 \text{ lb per hr}$$

The required rate of heat transfer can be calculated from this value, the specific heat, and the temperature drop of the air

$$q = 450 \times 0.24 \times (210 - 90) \\ = 13,000 \text{ Btu per hr}$$

The film coefficient  $h_1$  for the air can be determined by Cases 16 and 3 Since the internal cross-sectional area of  $1\frac{1}{4}$  in pipe is  $1.495 \text{ sq in}$ , the

weight velocity of the air

$$G = \frac{450 \times 144}{3,600 \times 1.495} = 12.0 \text{ lb.}/(\text{sec.})(\text{sq. ft.})$$

(Note that this value is larger than the minimum permissible value given in Table 16.) Then from Fig. 24,  $h_0 = 28 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.})$ ; from Table 14,  $F_t = 1.02$  (average temperature of the air =  $150^\circ\text{F.}$ ); and from Table 15,  $F_d = 0.94$  (inside pipe diameter = 1.38 in.). Hence, for the coil,

$$\begin{aligned} h_1 &= \left[ 1 + 3.54 \left( \frac{1.38}{24} \right) \right] \times 28 \times 1.02 \times 0.94 \\ &= 32 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.}). \end{aligned}$$

The film coefficient  $h_2$  for the water can be determined by Cases 17 and 11. The average pipe wall temperature  $t_w$  must first be determined by the method explained in Sec. 32: Assume  $t_w = 90^\circ\text{F.}$  Then, from Fig. 31,  $h_0 = 85 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.})$  [ $\Delta t = 90 - 80 = 10^\circ\text{F.}$ ]; from Table 26,  $F_t = 0.93$  [average film temperature =  $(90 + 80)/2 = 85^\circ\text{F.}$ ]; and from Table 27,  $F_d = 0.88$  (outside pipe diameter = 1.66 in.). Hence, as a first approximation,

$$\begin{aligned} h_2 &= 85 \times 0.93 \times 0.88 \\ &= 70 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.}). \end{aligned}$$

A more accurate value of  $t_w$  can now be determined by Eq. (45):

$$\begin{aligned} t_w &= \frac{32 \times 150 + 70 \times 80}{32 + 70} \\ &= 102^\circ\text{F.} \end{aligned}$$

Redetermining  $h_2$  for this value of  $t_w$ , from Fig. 31,

$$h_0 = 105 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.})$$

[ $\Delta t = 102 - 80 = 22^\circ\text{F.}$ ]; and from Table 26,

$$F_t = 0.96 \text{ [average film temperature} = \frac{(102 + 80)}{2} = 91^\circ\text{F.].}$$

Finally,

$$\begin{aligned} h_2 &= 105 \times 0.96 \times 0.88 \\ &= 89 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.}). \end{aligned}$$

The over-all coefficient  $U$ , based on the outside pipe surface, can be calculated by Eq. (27): Thus, since the pipe wall is 0.140 in. thick, and since the thermal conductivity of steel is about 26 B.t.u./(ft.)(hr.)( $^\circ\text{F.}$ ),

$$\frac{1}{U} = \frac{1.66}{32 \times 1.38} + \frac{0.140}{12 \times 26} + \frac{1}{89},$$

or

$$U = 20 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.}) \text{ (clean pipe).}$$

Allowance should be made for the presence of scale on the outside surface of the pipe. The scale coefficient  $h_s$  can be determined by Case 25: Assum-

ing that a deposit of scale 0.1 in. thick will gradually accumulate,

$$h_s = \frac{1 \times 12}{0.1} = 120 \text{ B t u / (sq ft) (hr) (°F)}$$

Then, by Eq. (29),

$$\frac{1}{U_s} = \frac{1}{20} + \frac{1}{120},$$

or

$$U_s = 17 \text{ B t u / (sq ft) (hr) (°F) (scale pipe)}$$

The mean temperature difference between the air and the water can be determined from Fig. 17 (or calculated by Eq. 30). Thus,

$$\Delta t = 210 - 80 = 130^\circ\text{F},$$

$$\Delta t'' = 90 - 80 = 10^\circ\text{F}, \text{ and } \Delta t_m = 47^\circ\text{F}$$

The outside pipe surface required can now be calculated by Eq. (24)

$$13,000 = 17 \times A \times 47,$$

or

$$A = 16.3 \text{ sq ft}$$

Since 2,301 lin. ft. of 1½ in. pipe have 1 sq. ft. of outside surface, the total length of pipe needed in the coil is  $2,301 \times 16.3 = 37,515$  ft.

**Illustrative Problem 6 (Evaporator)**—A weak salt solution is to be concentrated in a vertical short tube evaporator (see Sec. 40b). The evaporator will operate at a pressure of 24 in. Hg absolute and will be supplied with saturated steam at 10 lb per sq in. gauge. Assuming that the solution has approximately the same properties as water, determine the number of 2-in. o.d.  $\times$  No. 14 B.W.G. brass tubes 5 ft long required to evaporate 4,000 lb. of water per hour.

**Data**—Steam temperature =  $239^\circ\text{F}$ , solution temperature =  $201^\circ\text{F}$ , heat of condensation of the steam = 952 B t u per lb., heat supplied to the solution per lb. of water evaporated = 1,100 B t u.

**Solution**—The required rate of heat transfer

$$q = 4,000 \times 1,100 = 4,400,000 \text{ B t u per hr},$$

and the weight of steam required is  $4,400,000/952 = 4,620$  lb. per hr.

The film coefficient  $h_1$  for the solution (see, for water) can be determined by Case 24, and the film coefficient  $h_2$  for the steam can be determined by Case 22. Since both coefficients depend upon the tube-wall temperature  $t_w$ , a trial and error solution is necessary.

As a first trial, assume  $t_w = 225^\circ\text{F}$ . From Fig. 38,

$$h = 2200 \text{ B t u / (sq ft) (hr) (°F)} \quad (\Delta t = 225 - 201 = 24^\circ\text{F}),$$

and from Table 44,  $F_p = 0.94$  (pressure =  $24/30 = 0.8$  atm abs). Hence

$$\begin{aligned} h_1 &= 1.25 \times 2200 \times 0.94 \\ &= 2600 \text{ B t u / (sq ft) (hr) (°F)} \end{aligned}$$

The inside tube surface  $A_1$  corresponding to this value of  $h_1$  can be calculated by Eq. (23):

$$4,400,000 = 2,600 \times A_1 \times (225 - 201),$$

or

$$A_1 = 70.5 \text{ sq. ft.}$$

The corresponding outside tube surface (inside tube diameter = 1.834 in.) is  $70.5 \times (2.000/1.834) = 77.0$  sq. ft., and the rate of condensation of the steam  $W = 4,620/77.0 = 60.0$  lb./(sq. ft.)(hr.). Referring to Table 40, since  $W \times N = 60.0 \times 5 = 300$ , method *a* of Case 22 must be used to determine  $h_2$ . From Fig. 37,  $h_0 = 420$  B.t.u./(sq. ft.)(hr.)(°F.); from Table 41,  $F_t = 3.19$  (at  $t_w = 225^\circ\text{F.}$ ); and from Table 42,  $F_n' = 0.75$ . Hence, for clean tubes,  $h_2 = 420 \times 3.19 \times 0.75 = 1000$  B.t.u./(sq. ft.)(hr.)(°F.). Combining this value with a scale coefficient

$$h_2 = 3000 \text{ B.t.u./(sq. ft.)(hr.)(°F.) [from Case 25],}$$

$$\frac{1}{h_2'} = \frac{1}{1000} + \frac{1}{3000},$$

or

$$h_2' = 750 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

The tube-wall temperature  $t_w$  can now be calculated by Eq. (45):

$$\begin{aligned} t_w &= \frac{2600 \times 201 + 750 \times 239}{2600 + 750} \\ &= 209^\circ\text{F.} \end{aligned}$$

Since this value does not agree with the value initially assumed, the calculations must be repeated.

As a second trial, assume  $t_w = 220^\circ\text{F.}$  From Fig. 38,

$$h = 800 \text{ B.t.u./(sq. ft.)(hr.)(°F.)} \quad (\Delta t = 220 - 201 = 19^\circ\text{F.});$$

so

$$\begin{aligned} h_1 &= 1.25 \times 800 \times 0.94 \\ &= 940 \text{ B.t.u./(sq. ft.)(hr.)(°F.).} \end{aligned}$$

By Eq. (24),

$$4,400,000 = 940 \times A_1 \times (220 - 201)$$

or

$$A_1 = 246 \text{ sq. ft.}$$

The corresponding outside tube surface

$$A_2 = 246 \times \left( \frac{2.000}{1.834} \right) = 269 \text{ sq. ft.}$$

Hence,  $W = 4,620/269 = 17.2$  lb./(sq. ft.)(hr.). Since

$$W \times N = 17.2 \times 5 = 86,$$

method *b* of Case 22 must be used to determine  $h_2$ . From Fig. 36,

$$h_0 = 2000 \text{ B.t.u./(sq. ft.)(hr.)(°F.);}$$

from Table 37,  $F_1 = 1.56$  (at  $t_w = 220^\circ\text{F}$ ), and from Table 43,  $F_2'' = 1.27$ . Hence, for clean tubes,

$$h_1 = 0.29 \times 2,000 \times 1.56 \times 1.27 = 1150 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)}$$

Combining this with a scale coefficient  $h_s = 3000 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)}$ ,

$$\frac{1}{h_1'} = \frac{1}{1150} + \frac{1}{3000},$$

or

$$h_1' = 830 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)}$$

Calculating the tube-wall temperature  $t_w$  by Eq. (45),

$$\begin{aligned} t_w &= \frac{940 \times 201 + 830 \times 239}{940 + 830} \\ &= 219^\circ\text{F} \end{aligned}$$

Since this is approximately the value initially assumed, the calculations need not again be repeated.

The outside tube surface  $A_1 = 260 \text{ sq ft}$ , determined above, does not take into account the temperature drop through the tube wall. A more accurate value can be calculated as follows. Since the tube walls are 0.083 in thick and the thermal conductivity of brass at  $220^\circ\text{F}$  is about

$$59 \text{ Btu/(ft)(hr)(}^\circ\text{F.)},$$

by Eq. (27),

$$\frac{1}{U_1} = \frac{2.000}{940 \times 1.834} + \frac{0.083}{12 \times 59} + \frac{1}{830}$$

or

$$U_1 = 400 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)}$$

Hence, by Eq. (24),

$$4,400,000 = 400 \times A_1 \times (239 - 201),$$

or

$$A_1 = 290 \text{ sq ft}$$

Since 1910 lin ft of 2-in tubing have 1 sq ft of outside surface, the outside surface of each tube =  $5/1910 = 2.62 \text{ sq ft}$ , and the total number of tubes required =  $290/2.62 = 111$  tubes.

**Illustrative Problem 7**—Calculate the heat loss per foot of length from a bare 3-in steam pipe at  $400^\circ\text{F}$  if the surrounding air and walls are at  $70^\circ\text{F}$ .

**Solution**—The convection coefficient  $h$  can be determined by Case 12. Thus, from Fig. 32,

$$h_0 = 2.1 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)} \quad (\text{for } \Delta t = 400 - 70 = 330^\circ\text{F.)},$$

from Table 28,  $F_1 = 0.98$  [film temperature =  $(400 + 70)/2 = 235^\circ\text{F}$ ], from Table 29,  $F_2 = 0.73$  (outside pipe diameter = 3.500 in), and from Table 30,  $F_3 = 1.00$ . Hence,

$$\begin{aligned} h &= 2.1 \times 0.98 \times 0.73 \times 1.00 \\ &= 1.5 \text{ Btu/(sq ft)(hr)(}^\circ\text{F.)} \end{aligned}$$

The radiation coefficient  $h_r$  can be determined from Fig. 10 and is equal to 2.4 B.t.u./(sq. ft.)(hr.)(°F.). Assuming that the emissivity of the pipe is 0.9, the combined coefficient for convection and radiation is

$$1.5 + 0.9 \times 2.4 = 3.7 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(^\circ\text{F.}).$$

Therefore, since 1.091 ft. of 3-in. pipe have 1 sq. ft. of outside surface, the total heat loss from the pipe

$$= 3.7 \times \left( \frac{1}{1.091} \right) \times (400 - 70) = 1120 \text{ B.t.u.}/(\text{hr.})(\text{ft. of length}).$$



**Case 1. Liquids Heated inside Horizontal or Vertical Tubes, Turbulent Flow.**—The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_d$$

where  $h$  = the film coefficient, Btu/(sq ft)(hr)(°F);  $h_b$  = the base value of the film coefficient, from Fig 22,  $F_t$  = the temperature-correction factor, from Table 8,  $F_d$  = the diameter-correction factor, from Table 9

TABLE 8—TEMPERATURE-CORRECTION FACTOR FOR CASE 1

Liquid	Ave temp of liquid °F					
	0°F	50°F	100°F	150°F	200°F	250°F
Acetic acid (100%)			1.00	1.04	1.08	1.12
Acetic acid (50%)		0.75	1.00	1.15	1.30	1.45
Acetone	0.78	0.91	1.00	1.02	1.04	1.06
Ammonia	0.69	0.84	1.00	1.18	1.38	1.58
Amyl acetate	0.96	0.98	1.00	1.06	1.10	1.17
Amyl alcohol (iso)	0.41	0.68	1.00	1.39	1.82	2.25
Aniline		0.75	1.00	1.31	1.70	2.10
Benzene		0.80	1.00	1.14	1.28	1.42
Brine (CaCl <sub>2</sub> ) (25%)	0.54	0.74	1.00	1.29	1.63	2.01
Butyl alcohol (n)	0.59	0.73	1.00	1.33	1.72	2.15
Carbon disulfide	0.91	0.93	1.00	1.03	1.05	1.06
Carbon tetrachloride	0.75	0.88	1.00	1.05	1.09	1.12
Chlorobenzene	0.82	0.93	1.00	1.02	1.04	1.05
Chloroform	0.72	0.85	1.00	1.14	1.28	1.42
Ethyl acetate	1.01	1.01	1.00	0.99	0.98	0.97
Ethyl alcohol (100%)	0.63	0.80	1.00	1.21	1.43	1.64
Ethyl alcohol (90%)	0.58	0.64	1.00	1.41	1.80	2.40
Ethyl bromide	0.89	0.95	1.00	1.04	1.08	1.11
Ethylene glycol (50%)	0.45	0.66	1.00	1.40	1.89	2.40
Ethyl ether	0.82	0.94	1.00	1.06	1.12	1.18
Ethyl iodide	0.70	0.85	1.00	1.14	1.29	1.44
Glycerol (50%)	0.45	0.69	1.00	1.32	1.85	2.30
Heptane	0.89	0.92	1.00	1.08	1.18	1.29
Hexane	0.84	0.92	1.00	1.07	1.12	1.15
Methyl alcohol (100%)	0.69	0.87	1.00	1.10	1.19	1.28
Methyl alcohol (90%)	0.63	0.84	1.00	1.13	1.28	1.38
Methyl alcohol (40%)	0.59	0.67	1.00	1.30	1.61	1.91
Octane (n)	0.84	0.92	1.00	1.07	1.15	1.19
Pentane (n)	0.88	0.95	1.00	1.04	1.07	1.10
Propyl alcohol (iso)	0.58	0.69	1.00	1.32	1.64	1.95
Sulfur dioxide	0.96	0.98	1.00	1.03	1.07	1.11
Sulfuric acid (60%)		0.83	1.00	1.19	1.52	1.82
Toluene	0.80	0.90	1.00	1.05	1.16	1.23
Water		0.70	1.00	1.22	1.41	1.58

\* Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 9—DIAMETER-CORRECTION FACTOR FOR CASE 1

Inside tube dia in	$F_d$	Inside tube dia, in	$F_d$
0.20	1.38	1.20	0.97
0.30	1.27	1.30	0.95
0.40	1.20	1.40	0.94
0.50	1.15	1.50	0.92
0.60	1.11		
0.70	1.09	2.00	0.87
0.80	1.05	2.50	0.83
0.90	1.02	3.00	0.80
1.00	1.00	3.50	0.78
1.10	0.98	4.00	0.76

TABLE 10—MINIMUM PERMISSIBLE VALUES OF  $V \times d$  FOR CASE 1

$V$  = velocity, ft per sec  $d$  = inside tube dia in

Liquid*	Ave temp of liquid °F				
	50°F	100°F	150°F	200°F	250°F
	$(V \times d)_{\min}$				
Acetic acid (100%)	3.5	1.2	0.9	0.8	0.7
Acetic acid (50%)	3.5	1.7	1.0	0.7	0.6
Amyl acetate	1.6	1.0	0.6	0.4	0.3
Amyl alcohol (iso)	9.5	4.1	2.0	1.1	0.8
Aniline	7.6	3.7	1.9	1.0	0.7
Brine (CaCl <sub>2</sub> ) (25%)	1.9	1.1	0.7	0.4	0.2
Butyl alcohol (n)	6.2	3.2	1.7	0.9	0.6
Ethyl alcohol (100%)	2.4	1.5	1.0	0.6	0.5
Ethyl alcohol (90%)	5.8	2.3	1.2	0.8	0.7
Ethylene glycol (50%)	6.3	2.6	1.2	0.7	0.4
Glycerol (50%)	9.7	4.2	2.3	1.3	0.8
Methyl alcohol (40%)	3.6	1.5	0.8	0.6	0.4
Propyl alcohol (iso)	5.4	2.3	1.2	0.8	0.6
Sulfuric acid (60%)	6.5	3.4	2.2	1.8	1.7

\* Only those liquids for which  $(V \times d)_{\min}$  at 100°F is greater than 0.2 are listed

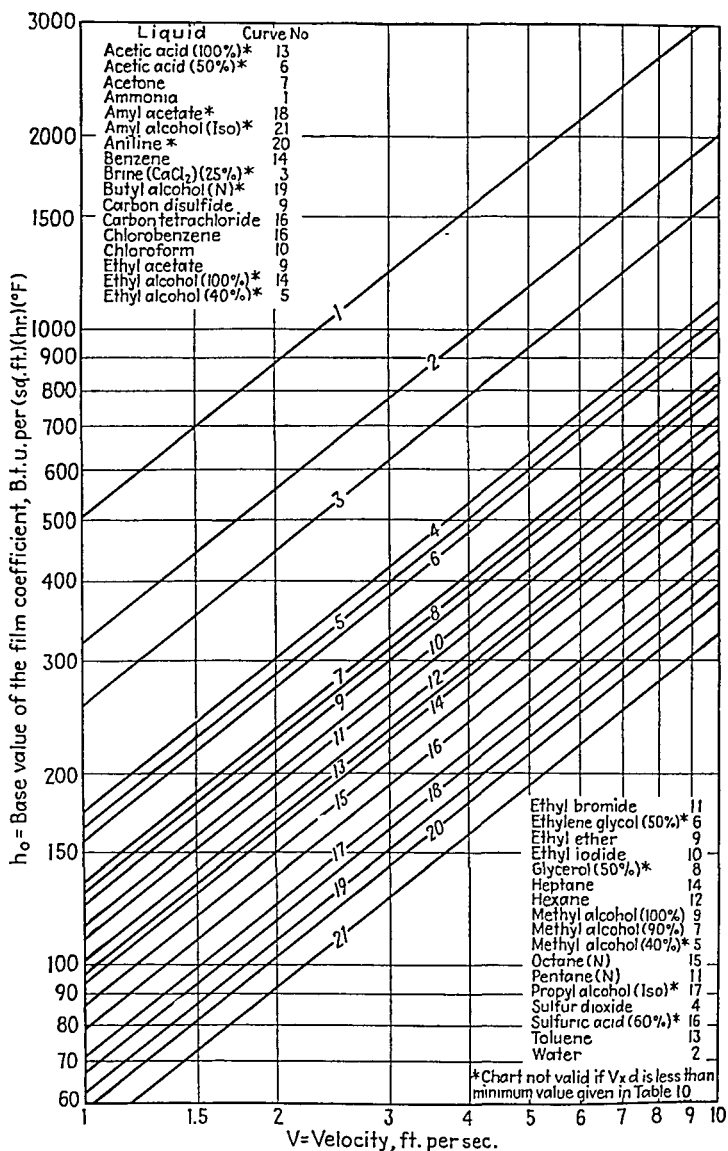


FIG. 22.—Base value of the film coefficient for Case 1.

**Case 2. Liquids Cooled inside Horizontal or Vertical Tubes, Turbulent Flow.**—The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_d$$

where  $h$  = the film coefficient, Btu/(sq ft)(hr)(°F),  $h_b$  = the base value of the film coefficient, from Fig. 23,  $F_t$  = the temperature-correction factor, from Table 11;  $F_d$  = the diameter-correction factor, from Table 12

TABLE 11—TEMPERATURE CORRECTION FACTOR FOR CASE 2

Liquid	Ave temp of liquid					
	0°F	50°F	100°F	150°F	200°F	250°F
	<i>F<sub>t</sub></i>					
Acetic acid (100%)			1.00	1.04	1.08	1.13
Acetic acid (50%)			1.00	1.26	1.47	1.65
Acetone	0.75	0.89	1.00	1.03	1.05	1.07
Ammonia	0.62	0.75	1.00	1.23	1.57	1.85
Amyl acetate	0.92	0.97	1.00	1.02	1.04	1.06
Amyl alcohol (iso)		0.63	1.00	1.47	2.01	2.55
Aniline		0.70	1.00	1.40	1.92	2.43
Benzene		0.78	1.00	1.18	1.35	1.53
Brine (CaCl <sub>2</sub> ) (25%)	0.48	0.70	1.00	1.33	1.83	2.35
Butyl alcohol (n)	0.42	0.69	1.00	1.40	1.88	2.35
Carbon disulfide	0.83	0.95	1.00	1.04	1.08	1.10
Carbon tetrachloride	0.60	0.83	1.00	1.07	1.13	1.17
Chlorobenzene	0.90	0.95	1.00	1.05	1.07	1.08
Chloroform	0.79	0.89	1.00	1.11	1.21	1.32
Ethyl acetate	0.89	1.00	1.00	1.00	0.99	0.98
Ethyl alcohol (100%)	0.58	0.77	1.00	1.25	1.52	1.80
Ethyl alcohol (40%)		0.58	1.00	1.45	1.90	2.35
Ethyl bromide	0.84	0.93	1.00	1.06	1.12	1.16
Ethylene glycol (50%)	0.37	0.60	1.00	1.53	2.18	
Ethyl ether	0.79	0.91	1.00	1.08	1.16	1.24
Ethyl iodide	0.69	0.85	1.00	1.15	1.30	1.45
Glycerol (50%)	0.37	0.63	1.00	1.45	1.90	2.35
Heptane	0.82	0.91	1.00	1.09	1.13	1.18
Hexane	0.81	0.90	1.00	1.09	1.17	1.23
Methyl alcohol (100%)	0.64	0.84	1.00	1.12	1.24	1.36
Methyl alcohol (90%)	0.61	0.82	1.00	1.14	1.2	1.39
Methyl alcohol (40%)	0.50	0.63	1.00	1.33	1.86	2.30
Octane (n)	0.79	0.90	1.00	1.09	1.16	1.24
Pentane (n)	0.86	0.93	1.00	1.05	1.10	1.13
Propyl alcohol (iso)		0.63	1.00	1.39	1.80	2.25
Sulfur dioxide	0.93	0.96	1.00	1.05	1.07	1.10
Sulfuric acid (60%)		0.65	1.00	1.23	1.56	1.85
Toluene	0.76	0.88	1.00	1.11	1.20	1.27
Water		0.66	1.00	1.30	1.56	1.77

\* Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 12—DIAMETER-CORRECTION FACTOR FOR CASE 2

Inside tube dia in	<i>F<sub>d</sub></i>	Inside tube dia in	<i>F<sub>d</sub></i>
0.20	1.33	1.20	0.97
0.30	1.27	1.30	0.95
0.40	1.20	1.40	0.94
0.50	1.15	1.50	0.92
0.60	1.11		
0.70	1.08	2.00	0.87
0.80	1.05	2.50	0.83
0.90	1.02	3.00	0.80
1.00	1.00	3.50	0.78
1.10	0.98	4.00	0.76

TABLE 13—MINIMUM PERMISSIBLE VALUES OF  $V \times d$  FOR CASE 2

$V$  = velocity, ft per sec,  $d$  = inside tube dia, in

Liquid*	Ave temp of liquid					
	50°F	100°F	150°F	200°F	250°F	
	$(V \times d)_{min}$					
Acetic acid (100%)		1.2	0.9	0.8	0.7	
Acetic acid (50%)	3.5	1.7	1.0	0.7	0.6	
Amyl acetate	1.8	1.0	0.6	0.4	0.3	
Amyl alcohol (iso)	9.5	4.1	2.0	1.1	0.8	
Aniline	7.6	3.7	1.9	1.0	0.7	
Brine (CaCl <sub>2</sub> ) (25%)	1.9	1.1	0.7	0.4	0.2	
Butyl alcohol (n)	6.2	3.2	1.7	0.9	0.6	
Ethyl alcohol (100%)	2.4	1.5	1.0	0.6	0.5	
Ethyl alcohol (40%)	5.8	2.3	1.2	0.8	0.7	
Ethylene glycol (50%)	6.3	2.6	1.2	0.7	0.4	
Glycerol (50%)	9.7	4.2	2.3	1.3	0.8	
Methyl alcohol (40%)	3.6	1.5	0.8	0.6	0.4	
Propyl alcohol (iso)	5.4	2.3	1.2	0.8	0.6	
Sulfuric acid (60%)	6.5	3.4	2.2	1.8	1.7	

\* Only those liquids for which  $(V \times d)_{min}$  at 100°F is greater than 0.2 are listed

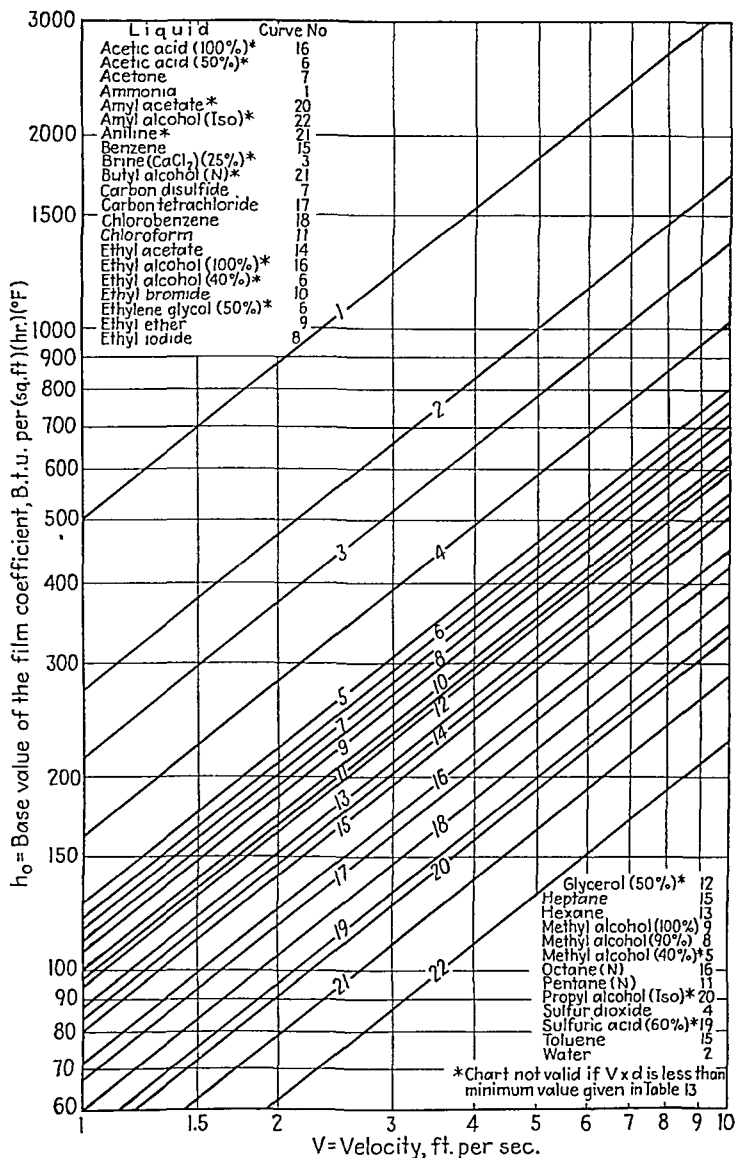


FIG. 23.—Base value of the film coefficient for Case 2.

**Case 3. Gases Heated or Cooled inside Horizontal or Vertical Tubes, Turbulent Flow.**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d,$$

where  $h$  = the film coefficient, B t u / (sq ft) (hr) (°F)

$h_0$  = the base value of the film coefficient, from Fig 24

$F_t$  = the temperature-correction factor, from Table 14

$F_d$  = the diameter-correction factor, from Table 15

TABLE 14—TEMPERATURE-CORRECTION FACTOR FOR CASE 3

Gas	Ave temp of gas						
	-100°F	0°F	100°F	200°F	300°F	400°F	500°F
	$F_t^*$						
Acetone			1.00	1.15	1.32	1.59	1.83
Acetylene	0.84	0.92	1.00	1.08	1.15	1.23	1.22
Air	0.90	0.98	1.00	1.04	1.07	1.09	1.12
Ammonia	0.78	0.91	1.00	1.07	1.15	1.18	1.22
Benzene			1.00	1.20	1.44	1.69	1.94
Butane		0.84	1.00	1.09	1.17	1.23	1.28
Carbon dioxide	0.82	0.92	1.00	1.06	1.11	1.16	1.22
Carbon monoxide	0.83	0.93	1.00	1.03	1.09	1.13	1.17
Chlorine	0.92	0.96	1.00	1.03	1.06	1.09	1.11
Chloroform			1.00	1.09	1.19	1.29	1.38
Ethane	0.73	0.87	1.00	1.13	1.28	1.39	1.52
Ethyl acetate			1.00	1.18	1.35	1.51	1.66
Ethyl alcohol			1.00	1.04	1.08	1.12	1.16
Ethyl chloride		0.88	1.00	1.07	1.11	1.15	1.18
Ethylene	0.75	0.88	1.00	1.11	1.22	1.33	1.44
Ethyl ether		0.89	1.00	1.12	1.26	1.41	1.56
Helium	0.94	0.97	1.00	1.05	1.06	1.09	1.12
Hydrogen	0.91	0.96	1.00	1.04	1.08	1.12	1.16
Hydrogen sulfide	0.85	0.92	1.00	1.07	1.14	1.20	1.26
Methane	0.83	0.93	1.00	1.05	1.09	1.13	1.16
Methyl chloride	0.68	0.83	1.00	1.16	1.31	1.46	1.60
Nitric oxide	0.91	0.96	1.00	1.03	1.06	1.08	1.10
Nitrogen	0.93	0.96	1.00	1.03	1.05	1.07	1.09
Nitrous oxide	0.96	0.98	1.00	1.02	1.04	1.06	1.08
Oxygen	0.91	0.96	1.00	1.03	1.06	1.08	1.10
Pentane (iso)		0.88	1.00	1.14	1.30	1.49	1.74
Steam			1.00	1.03	1.06	1.10	1.14
Sulfur dioxide		0.94	1.00	1.05	1.10	1.14	1.18

\* Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 15—DIAMETER CORRECTION FACTOR FOR CASE 3

Inside Tube Dia. In.	$F_d$
0.20	1.38
0.30	1.27
0.40	1.20
0.50	1.15
0.60	1.11
0.70	1.08
0.80	1.05
0.90	1.02
1.00	1.00
1.10	0.98
1.20	0.97
1.30	0.95
1.40	0.94
1.50	0.92
2.00	0.87
2.50	0.83
3.00	0.80
3.50	0.78
4.00	0.76

TABLE 16—MINIMUM PERMISSIBLE VALUES OF  $G = V\rho$  FOR CASE 3

$G = \frac{\text{rate of flow}}{\text{cross-sectional area}}$

$\frac{\text{lb per sec}}{\text{sq ft}}$

$V = \text{velocity, ft per sec}$

$\rho = \text{density, lb per cu ft}$

$G_{\min} = (V\rho)_{\min}$

Pressure Lb per Sq In Gauge	$G_{\min}$
0	0.46
50	1.19
100	1.73
150	2.18
200	2.59
250	2.97
300	3.32

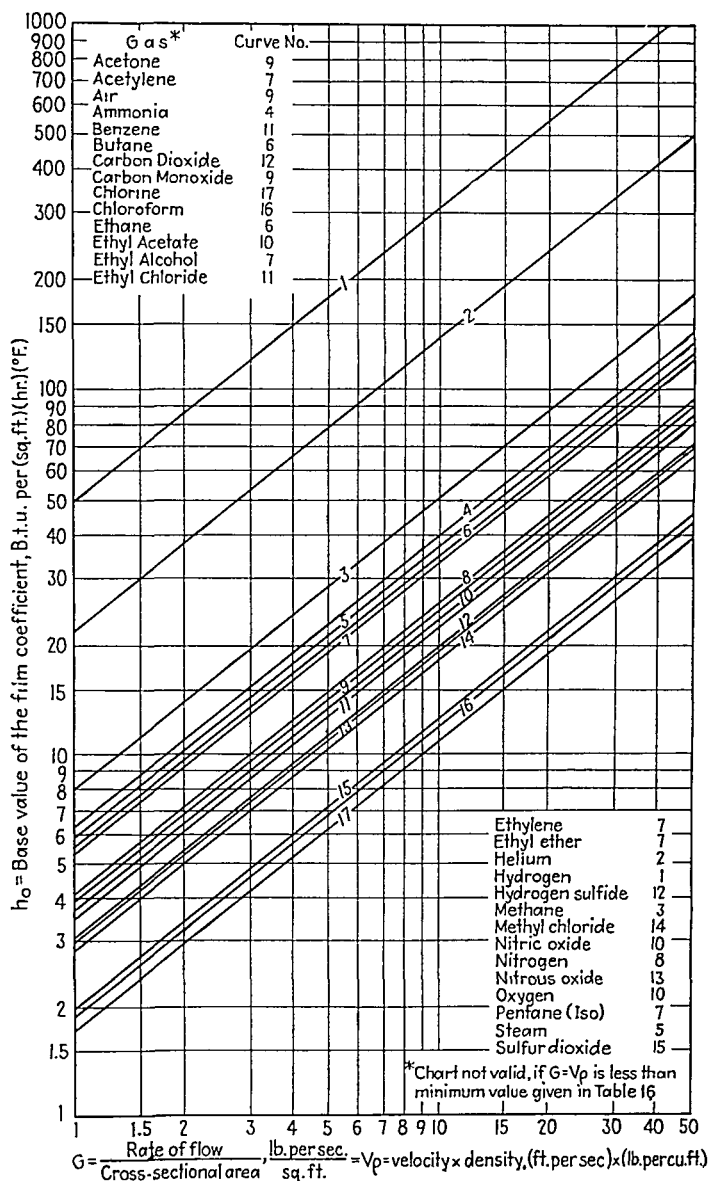


FIG. 24.—Base value of the film coefficient for Case 3.

**Case 4. Gases at Atmospheric Pressure Heated or Cooled inside Horizontal or Vertical Tubes, Turbulent Flow**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d$$

where  $h$  = the film coefficient, Btu/(sq ft)(hr)(°F)

$h_0$  = the base value of the film coefficient, from Fig. 25

$F_t$  = the temperature-correction factor, from Table 17

$F_d$  = the diameter-correction factor, from Table 18.

TABLE 17—TEMPERATURE-CORRECTION FACTOR FOR CASE 4

Gas	Ave temp of gas						
	-100°F	0°F	100°F	200°F	300°F	400°F	500°F
	$F_t^*$						
Acetylene	1.26	1.12	1.00	0.91	0.84	0.76	0.69
Air	1.29	1.14	1.00	0.91	0.84	0.78	0.73
Ammonia		1.05	1.00	0.94	0.87	0.80	0.72
Butane			1.00	0.96	0.93	0.81	0.89
Carbon dioxide	1.17	1.08	1.00	0.93	0.87	0.83	0.79
Carbon monoxide	1.27	1.12	1.00	0.89	0.81	0.73	0.63
Chlorine		1.17	1.00	0.91	0.85	0.74	0.66
Ethane	1.04	1.07	1.00	0.98	0.97	0.96	0.95
Ethylene	1.07	1.03	1.00	0.95	0.97	0.96	0.95
Helium	1.33	1.14	1.00	0.89	0.90	0.71	0.63
Hydrogen	1.30	1.13	1.00	0.90	0.81	0.72	0.64
Hydrogen sulfide		1.09	1.00	0.94	0.89	0.85	0.81
Methane	1.19	1.09	1.00	0.97	0.84	0.76	0.69
Methyl chloride		0.97	1.00	1.02	1.03	1.04	1.05
Nitric oxide	1.30	1.17	1.00	0.90	0.82	0.74	0.67
Nitrogen	1.30	1.13	1.00	0.90	0.81	0.72	0.64
Nitrous oxide	1.36	1.14	1.00	0.88	0.79	0.69	0.60
Oxygen	1.30	1.12	1.00	0.90	0.87	0.74	0.67
Sulfur dioxide			1.00	0.92	0.86	0.81	0.77

\* Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 18—DIAMETER CORRECTION FACTOR FOR CASE 4

Inside Tube Dia., In	$F_d$
0.20	1.38
0.30	1.27
0.40	1.20
0.50	1.15
0.60	1.11
0.70	1.08
0.80	1.05
0.90	1.02
1.00	1.00
1.10	0.98
1.20	0.97
1.30	0.95
1.40	0.94
1.50	0.92
2.00	0.87
2.50	0.83
3.00	0.80
3.50	0.78
4.00	0.76

TABLE 19—MINIMUM PERMISSIBLE VALUES OF  $V$  FOR CASE 4

$V$  = velocity ft per sec

Gas*	Ave temp of gas					
	0°F	100°F	200°F	300°F	400°F	500°F
	$V_{min}$					
Ammonia	9	11	13	15	17	19
Helium	39	47	55	64	72	81
Hydrogen	75	97	108	125	141	157
Methane	10	12	14	16	18	20

\* Only those gases for which  $V_{min}$  is greater than 10 ft per sec. at temperatures up to 500°F are listed.

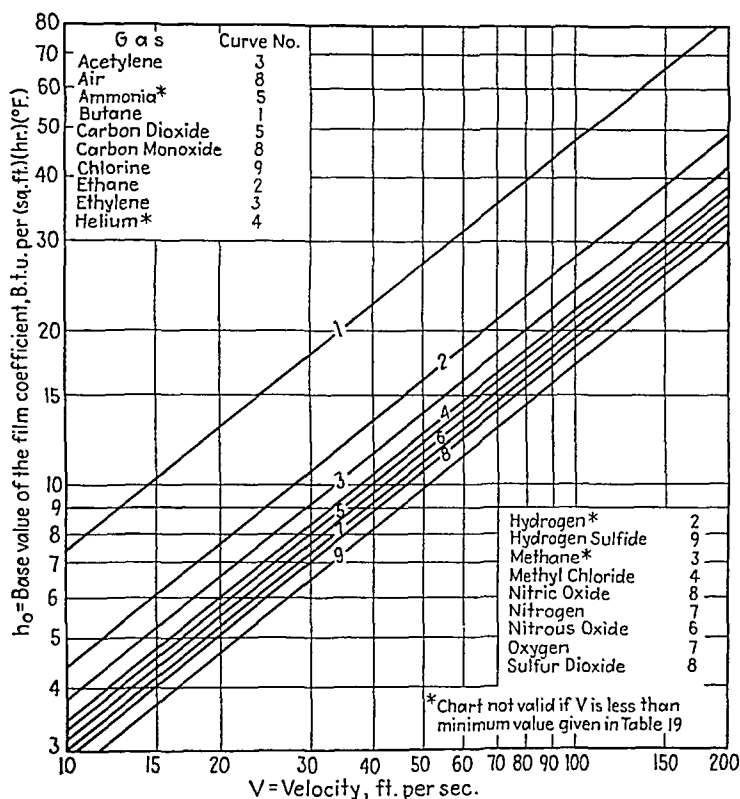


FIG. 25.—Base value of the film coefficient for Case 4.



Case 5 Liquids Heated or Cooled outside Single Tubes, Direction of Flow Normal to Tube—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d$$

where  $h$  = the film coefficient, B t u / (sq ft) (hr) (°F)

$h_0$  = the base value of the film coefficient from Fig. 26

$F_t$  = the temperature-correction factor, from Table 20

$F_d$  = the diameter-correction factor, from Table 21

TABLE 20—TEMPERATURE-CORRECTION FACTOR FOR CASE 5

Liquid	Ave temp of the film*					
	0°F	50°F	100°F	150°F	200°F	250°F
	$F_t$ †					
Acetic acid (100%)			1.00	0.98	0.92	0.88
Acetic acid (50%)		0.82	1.00	1.12	1.19	1.23
Acetone	0.80	0.95	1.00	1.01	1.02	1.03
Ammonia	0.79	0.89	1.00	1.11	1.23	1.31
Amyl acetate	1.16	1.08	1.00	0.93	0.86	0.78
Amyl alcohol (iso)	0.57	0.77	1.00	1.24	1.48	1.72
Aniline		0.84	1.00	1.20	1.42	1.68
Benzene		0.85	1.00	1.08	1.16	1.24
Brine (CaCl <sub>2</sub> ) (25%)	0.63	0.80	1.00	1.21	1.47	1.70
Butyl alcohol (n)	0.83	0.89	1.00	1.21	1.48	1.85
Carbon disulfide	0.97	0.98	1.00	1.01	1.02	1.02
Carbon tetrachloride		0.92	1.00	1.01	1.02	1.03
Chlorobenzene	1.06	1.03	1.00	0.97	0.95	0.94
Chloroform	0.88	0.94	1.00	1.00	1.00	1.13
Ethyl acetate	1.12	1.06	1.00	0.94	0.87	0.81
Ethyl alcohol (100%)	0.74	0.87	1.00	1.13	1.25	1.36
Ethyl alcohol (40%)	0.47	0.72	1.00	1.28	1.65	1.88
Ethyl bromide	0.83	0.97	1.00	1.02	1.03	1.04
Ethylene glycol (50%)	0.62	0.74	1.00	1.28	1.58	1.94
Ethyl ether	0.91	0.96	1.00	1.00	1.09	1.15
Ethyl iodide	0.78	0.89	1.00	1.10	1.19	1.27
Glycerol (50%)	0.69	0.77	1.00	1.55	1.73	
Heptane	0.91	0.96	1.00	1.03	1.06	1.09
Hexane	0.91	0.95	1.00	1.04	1.07	1.10
Methyl alcohol (100%)	0.79	0.91	1.00	1.06	1.10	1.15
Methyl alcohol (90%)	0.76	0.89	1.00	1.08	1.14	1.20
Methyl alcohol (40%)	0.60	0.76	1.00	1.20	1.36	1.50
Octane (n)	0.91	0.96	1.00	1.00	1.09	1.15
Pentane (n)	0.94	0.97	1.00	1.02	1.05	1.04
Propyl alcohol (iso)	0.53	0.77	1.00	1.21	1.38	1.62
Sulfur dioxide	1.03	1.01	1.00	0.99	0.99	0.98
Sulfuric acid (60%)		0.80	1.00	1.09	1.19	1.28
Toluene	0.90	0.90	1.00	1.04	1.07	1.09
Water		0.77	1.00	1.16	1.30	1.41

\* Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 21—DIAMETER CORRECTION FACTOR FOR CASE 5

Outside Tube D <sub>o</sub> In	$F_d$
0.250	1.84
0.375	1.54
0.500	1.36
0.625	1.23
0.750	1.14
1.00	1.00
1.25	0.91
1.50	0.84
2.00	0.74
2.50	0.67
3.00	0.62
3.50	0.58
4.00	0.54

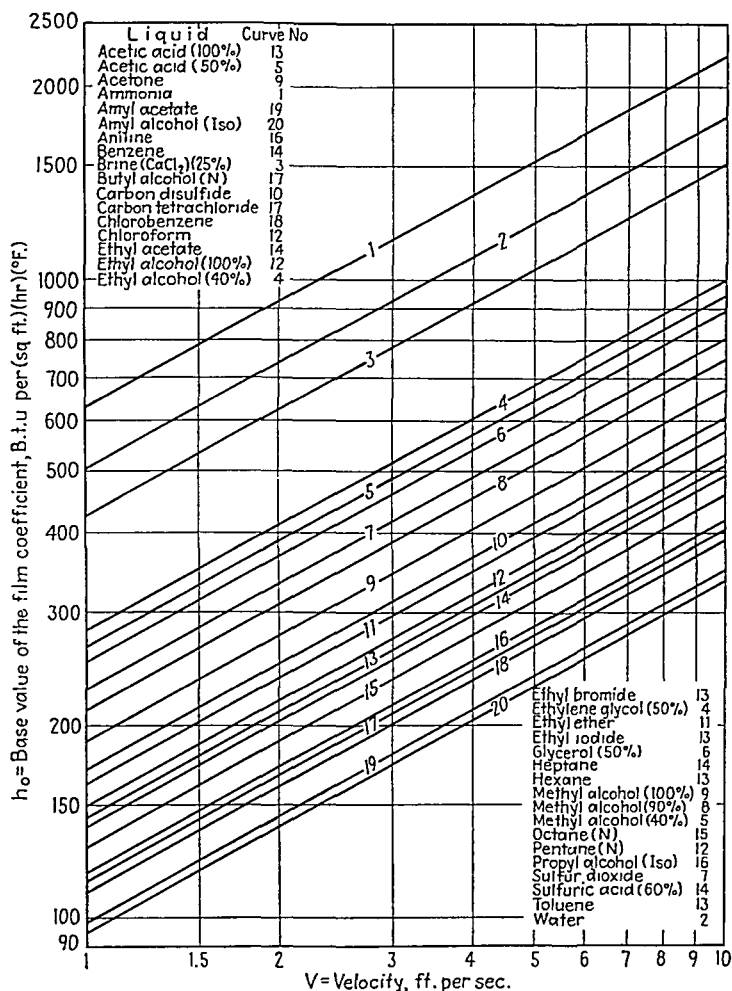


FIG. 26.—Base value of the film coefficient for Case 5.

Case 7. Gases at Atmospheric Pressure Heated or Cooled outside Single Tubes, Direction of Flow Normal to Tube —The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d$$

where  $h$  = the film coefficient, B t u / (sq ft) (hr) (°F)

$h_0$  = the base value of the film coefficient, from Fig. 28

$F_t$  = the temperature-correction factor, from Table 24

$F_d$  = the diameter correction factor from Table 25

Note that the film coefficient  $h$  obtained from the foregoing equation is for heat transfer by convection only. If the heat-transfer surface is exposed to other surfaces at different temperatures, it will gain or lose heat by radiation also. The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IV in the Appendix, times the radiation coefficient  $h_r$ , obtained from Fig. 10 in Chap. II, onto the convection coefficient  $h$  obtained from the equation above.

TABLE 24 —TEMPERATURE CORRECTION FACTOR FOR CASE 7

Gas	Ave temp of the film*						
	-100°F	0°F	100°F	200°F	300°F	400°F	500°F
	$F_t$ †						
Acetylene		0.97	1.00	1.03	1.05	1.07	1.09
Air	1.07	1.03	1.00	0.97	0.94	0.91	0.90
Ammonia		0.94	1.00	1.01	1.02	1.03	1.04
Butane			1.00	1.01	1.10	1.13	1.16
Carbon dioxide	0.94	0.98	1.00	1.01	1.01	1.02	1.02
Carbon monoxide	1.03	1.02	1.00	0.98	0.97	0.95	0.93
Chlorine		1.03	1.00	0.98	0.96	0.94	0.93
Ethane	0.84	0.93	1.00	1.08	1.16	1.24	1.33
Ethylene	0.89	0.95	1.00	1.06	1.12	1.18	1.25
Helium	1.13	1.06	1.00	0.95	0.90	0.86	0.82
Hydrogen	1.03	1.04	1.00	0.96	0.93	0.90	0.86
Hydrogen sulfide		1.01	1.00	0.98	0.97	0.95	0.93
Methane	1.09	1.04	1.00	0.97	0.95	0.92	0.90
Methyl chloride		0.87	1.00	1.11	1.20	1.29	1.38
Nitric oxide	1.06	1.03	1.00	0.97	0.95	0.93	0.90
Nitrogen	1.09	1.04	1.00	0.96	0.93	0.90	0.86
Nitrous oxide	1.19	1.08	1.00	0.94	0.88	0.83	0.79
Oxygen	1.07	1.03	1.00	0.97	0.95	0.91	0.89
Sulfur dioxide			1.00	1.00	0.99	0.98	0.98

\* Average temperature of the film = (temperature of the tube wall + temperature of the gas) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 25 —DIAMETER-CORRECTION FACTOR FOR CASE 7

Outside Tube	$F_d$
$D_o \text{ In}$	
0.250	1.84
0.375	1.54
0.500	1.36
0.625	1.23
0.750	1.14
1.00	1.00
1.25	0.91
1.50	0.84
2.00	0.74
2.50	0.67
3.00	0.62
3.50	0.58
4.00	0.54

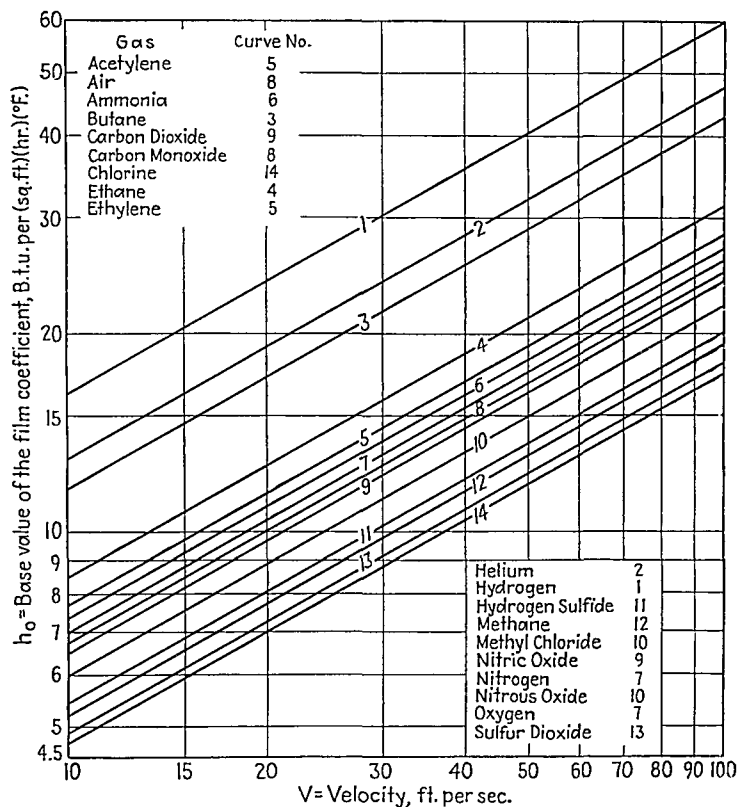


FIG. 28.—Base value of the film coefficient for Case 7.

**Case 8 Liquids or Gases Heated or Cooled outside Tube Bundles, Direction of Flow Normal to Tubes**—Approximate values of the film coefficients for fluids flowing outside tube bundles and in a direction normal to the tubes can be determined by the following equations  $h = 1.3 \times h_{s,1}$  for *staggered* tubes or  $h = 1.2 \times h_{s,1}$  for tubes *in line* where  $h$  = the film coefficient for flow normal to the tube bundle, B t u / (sq ft) (hr) (°F), and  $h_{s,1}$  = the film coefficient for flow normal to a single tube, obtained from Case 5, 6, or 7.  $h_{s,1}$  should be evaluated at a velocity equal to that at the narrowest section between the tubes of the tube bundle.

For baffled heat exchangers in which the flow is normal to most of the tubes but parallel to some it is suggested that the film coefficient for the shell side fluid be taken equal to  $h_{s,1}$ , the latter being evaluated at the velocity

$$V = \frac{144Q}{(d_s - nd)l}$$

where  $V$  = the velocity between the tubes at the widest part of the shell, ft per sec,  $Q$  = the volumetric rate of flow cu ft per sec,  $d_s$  = the inside diameter of the shell in,  $d$  = the outside diameter of the tubes in,  $n$  = the number of tubes across the widest part of the shell,  $l$  = the distance between baffles in.

**Case 9 Liquids or Gases Heated or Cooled outside Tube Bundles, Direction of Flow Parallel to Tubes**—The film coefficient for fluids flowing *outside* tube bundles and in a direction parallel to the tubes can be determined from Case 1, 2, 3, or 4 for fluids *inside* tubes if an equivalent inside diameter  $d_e$  is used in determining the diameter correction factor  $F_d$ . The equivalent diameter can be calculated by the equation

$$d_e = 4 \times \frac{A}{P} \quad (46)$$

where  $d_e$  = the equivalent inside diameter in,  $A$  = the cross sectional area of the space between the tubes sq in,  $P$  = the number of tubes  $\times$  the perimeter of each tube in.

For tubes spaced on equilateral triangles or on squares  $d_e$  can be determined from Fig. 29 or 30.

**Case 10 Liquids or Gases Heated or Cooled in Annular Spaces, Turbulent Flow**—The film coefficients for fluids flowing through the *annular space* of a double-pipe heat exchanger can be determined from Case 1, 2, 3, or 4 for fluids *inside* tubes if an equivalent inside diameter  $d_e$  is used in determining the diameter correction factor  $F_d$ . The equivalent diameter can be calculated by the equation

$$d_e = \frac{d_1^3 - d_2^3}{d_2} \quad (47)$$

where  $d_e$  = the equivalent inside diameter in,  $d_1$  = the inside diameter of the outer pipe in,  $d_2$  = the outside diameter of the inner pipe in.

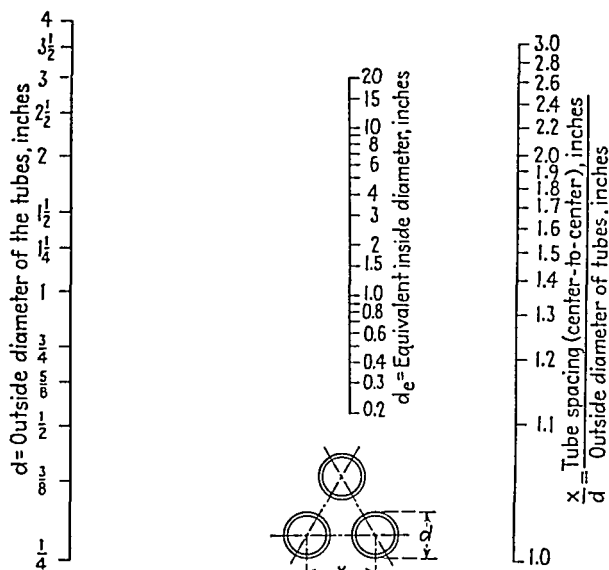


FIG. 29.—Alignment chart for determining the equivalent inside diameter for parallel-flow outside tube bundles, tubes spaced on equilateral triangles.

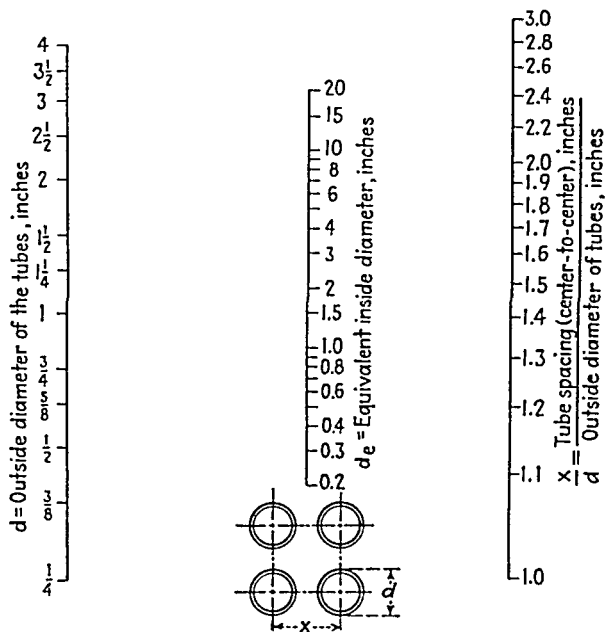


FIG. 30.—Alignment chart for determining the equivalent inside diameter for parallel flow outside tube bundles, tubes spaced on squares.

Case 11. Liquids Heated outside Single Horizontal Tubes, Natural Convection.—The film coefficient can be determined by the equation

$$h = h_0 \times F_1 \times F_d,$$

where  $h$  = the film coefficient, Btu/(sq ft)(hr)(°F)

$h_0$  = the base value of the film coefficient, from Fig 31

$F_1$  = the temperature-correction factor, from Table 26

$F_d$  = the diameter correction factor, from Table 27

TABLE 26—TEMPERATURE CORRECTION FACTOR FOR CASE 11

Liquid	Ave temp of the film*					
	0°F	50°F	100°F	150°F	200°F	250°F
	$F_1$ †					
Acetic acid (100%)			1.00	0.95	0.80	0.87
Acetone		0.97	1.00	1.01	1.02	1.03
Ammonia	0.79	0.89	1.00	1.11	1.22	1.33
Benzene		0.91	1.00	1.08	1.16	1.23
Carbon disulfide	0.97	0.98	1.00	1.01	1.02	1.03
Carbon tetrachloride		0.93	1.00	0.99	0.98	0.99
Chlorobenzene	1.11	1.05	1.00	0.95	0.91	0.88
Chloroform	0.83	0.95	1.00	1.05	1.09	1.14
Ethyl acetate	1.17	1.08	1.00	0.92	0.85	0.78
Ethyl alcohol (100%)	0.76	0.88	1.00	1.12	1.23	1.33
Ethyl alcohol (40%)	0.49	0.73	1.00	1.29	1.58	
Ethyl bromide	0.93	0.98	1.00	1.02	1.03	1.04
Ethyl ether	0.90	0.96	1.00	1.03	1.06	1.08
Ethyl iodide	0.80	0.90	1.00	1.08	1.16	1.24
Heptane	0.93	0.98	1.00	1.03	1.05	1.07
Hexane	0.89	0.95	1.00	1.04	1.07	1.09
Methyl alcohol (100%)	0.83	0.91	1.00	1.06	1.10	1.14
Methyl alcohol (90%)	0.74	0.89	1.00	1.09	1.17	1.25
Octane (n)	0.92	0.96	1.00	1.04	1.08	1.11
Pentane (n)	0.96	0.98	1.00	1.02	1.05	1.07
Sulfur dioxide	1.04	1.02	1.00	0.95	0.98	0.97
Sulfuric acid (98%)		0.76	1.00	1.24	1.48	
Sulfuric acid (80%)		0.80	1.00	1.17	1.30	1.38
Toluene	0.90	0.95	1.00	1.04	1.07	1.08
Water		0.77	1.00	1.15	1.26	1.30

\* Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 27—DIAMETER-CORRECTION FACTOR FOR CASE 11

Outside Tube	$F_d$
Dia. In	
0.250	1.41
0.375	1.28
0.500	1.19
0.625	1.13
0.750	1.08
1.00	1.00
1.25	0.93
1.50	0.90
2.00	0.84
2.50	0.80
3.00	0.76
3.50	0.73
4.00	0.71

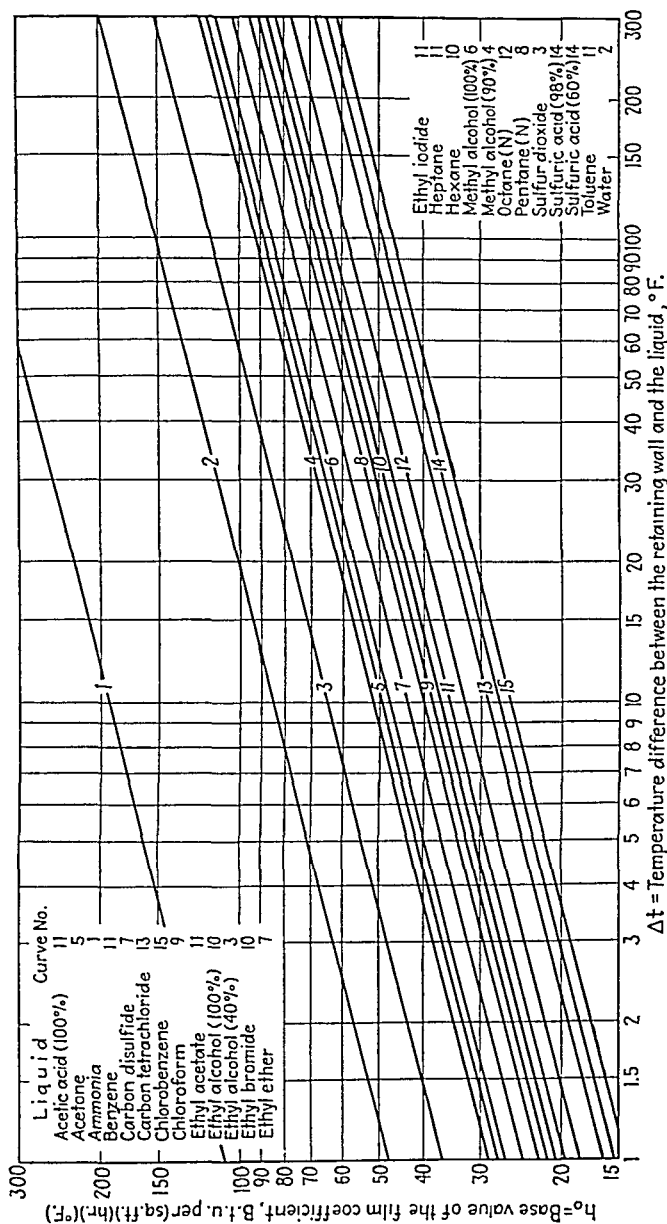


FIG. 31.—Base value of the film coefficient for Case 11.



**Case 12 Gases Heated outside Single Horizontal Tubes, Natural Convection**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d \times F_p$$

where  $h$  = the film coefficient, Btu/(sq ft)(hr)(°F),  $h_0$  = the base value of the film coefficient, from Fig 32,  $F_t$  = the temperature-correction factor, from Table 28,  $F_d$  = the diameter-correction factor, from Table 29,  $F_p$  = the pressure-correction factor, from Table 30

Note that the film coefficient  $h$  obtained from the foregoing equation is for heat transfer by convection only. If the heat-transfer surface is exposed to other surfaces at different temperatures it will gain or lose heat by radiation also. The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IV in the Appendix times the radiation coefficient  $h_r$ , obtained from Fig 10 in Chap II, onto the convection coefficient  $h$  obtained from the equation above.

TABLE 28—TEMPERATURE-CORRECTION FACTOR FOR CASE 12

Gas	Ave temp of the film*						
	-100°F	0°F	100°F	200°F	300°F	400°F	500°F
	F †						
Acetone			1.00	1.12	1.27	1.41	1.55
Acetylene	0.87	0.94	1.00	1.06	1.12	1.18	1.23
Air	1.03	1.01	1.00	0.99	0.97	0.96	0.95
Ammonia	0.87	0.94	1.00	1.05	1.10	1.15	1.19
Benzene			1.00	1.17	1.36	1.58	1.80
Butane		0.92	1.00	1.08	1.16	1.24	1.31
Carbon dioxide	0.88	0.95	1.00	1.03	1.05	1.07	1.09
Carbon monoxide	1.00	1.00	1.00	0.99	0.98	0.97	0.96
Chlorine		1.01	1.00	0.99	0.98	0.98	0.97
Chloroform			1.00	1.07	1.14	1.20	1.27
Ethane	0.80	0.90	1.00	1.10	1.19	1.28	1.37
Ethyl acetate			1.00	1.15	1.29	1.43	1.57
Ethyl alcohol			1.00	0.99	0.97	0.96	0.94
Ethyl chloride			1.00	1.06	1.12	1.17	1.22
Ethylene	0.83	0.91	1.00	1.09	1.17	1.25	1.33
Ethyl ether			1.00	1.11	1.21	1.32	1.43
Helium	1.07	1.03	1.00	0.98	0.97		
Hydrogen	1.04	1.02	1.00	0.98	0.96	0.94	0.92
Hydrogen sulfide		0.99	1.00	1.01	1.01		
Methane	0.83	0.95	1.00	1.05	1.10	1.15	1.19
Methyl chloride		0.86	1.00	1.12	1.23	1.34	1.45
Nitric oxide	1.01	1.01	1.00	0.99	0.97	0.95	0.93
Nitrogen	1.03	1.02	1.00	0.98	0.96	0.94	0.91
Nitrous oxide	1.15	1.06	1.00	0.95	0.91	0.87	
Oxygen	1.02	1.01	1.00	0.99	0.97	0.95	0.93
Pentane (so)		0.90	1.00	1.11	1.24	1.41	1.53
Steam			1.00	1.02	1.04	1.06	1.08
Sulfur dioxide			1.00	1.02	1.03	1.04	1.04

\* Average temperature of the film = (temperature of the tube wall + temperature of the gas in body of the gas) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 29—DIAMETER CORRECTION FACTOR FOR CASE 12

Outside tube dia in	<i>F<sub>d</sub></i>	Outside tube dia in	<i>F<sub>d</sub></i>
0.250	1.41	1.50	0.90
0.375	1.28	2.00	0.84
0.500	1.19	2.50	0.80
0.625	1.13	3.00	0.76
0.750	1.08	3.50	0.73
1.00	1.00	4.00	0.71
1.25	0.93		

TABLE 30—PRESSURE-CORRECTION FACTOR FOR CASE 12

Pressure lb per sq in gauge	<i>F<sub>p</sub></i>	Pressure lb per sq in gauge	<i>F<sub>p</sub></i>
0	1.00	100	2.79
10	1.30	150	3.3
20	1.54	200	3.8
30	1.74	250	4.2
40	1.93	300	4.6
50	2.10	350	5.0
60	2.26	400	5.3
70	2.40	450	5.6
80	2.54	500	5.9
90	2.67		

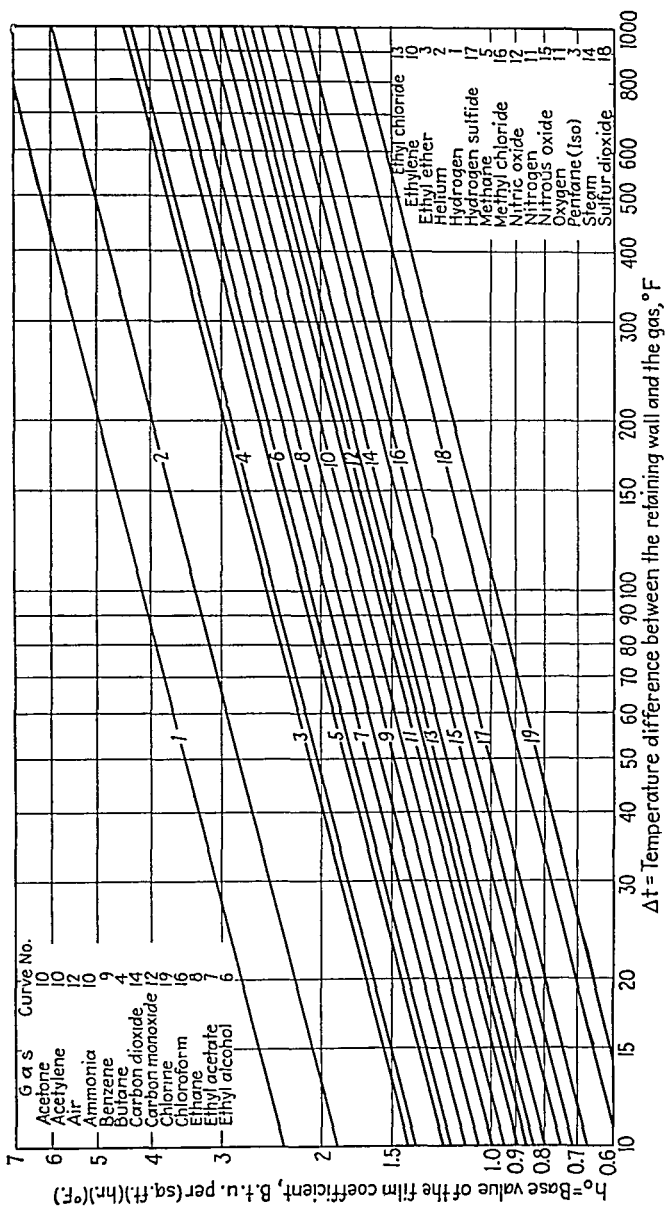


FIG. 32.—Base value of the film coefficient for Case 12.

**Case 13 Liquids Heated inside or outside Vertical Tubes or on Vertical Plates, Low Velocities or Natural Convection Only**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t$$

where  $h$  = the film coefficient Btu/(sq ft)(hr)(°F)

$h_0$  = the base value of the film coefficient from Fig 33

$F_t$  = the temperature-correction factor from Table 31

TABLE 31—TEMPERATURE-CORRECTION FACTOR FOR CASE 13

Liquid	Ave temp of the film*					
	0°F	50°F	100°F	150°F	200°F	250°F
	$F_t$ †					
Acetic acid (100%)			1.00	0.99	0.99	0.99
Acetone	0.87	0.94	1.00	1.03	1.05	1.07
Ammonia	0.73	0.86	1.00	1.15	1.32	1.51
Benzene		0.88	1.00	1.11	1.22	1.33
Carbon disulfide	0.94	0.97	1.00	1.03	1.06	1.09
Carbon tetrachloride	0.83	0.92	1.00	1.03	1.05	1.07
Chlorobenzene	1.00	1.00	1.00	0.99	0.99	0.98
Chloroform	0.85	0.93	1.00	1.07	1.13	1.20
Ethyl acetate	1.0	1.04	1.00	0.96	0.92	0.87
Ethyl alcohol (100%)	0.68	0.83	1.00	1.17	1.33	1.50
Ethyl alcohol (40%)	0.40	0.68	1.00	1.3	1.77	
Ethyl bromide	0.91	0.96	1.00	1.03	1.05	1.0
Ethyl ether	0.87	0.91	1.00	1.05	1.08	1.10
Ethyl iodide	0.1	0.87	1.00	1.12	1.23	1.35
Heptane	0.89	0.94	1.00	1.06	1.11	1.1
Hexane	0.88	0.94	1.00	1.06	1.09	1.12
Methyl alcohol (100%)	0.76	0.89	1.00	1.09	1.18	1.25
Methyl alcohol (90%)	0.6	0.86	1.00	1.12	1.23	1.34
Octane (n)	0.86	0.94	1.00	1.07	1.13	1.20
Pentane (n)	0.91	0.96	1.00	1.03	1.05	1.07
Sulfur dioxide	1.00	1.00	1.00	1.00	1.00	1.00
Sulfuric acid (98%)		0.76	1.00	1.23	1.46	1.69
Sulfuric acid (60%)		0.76	1.00	1.19	1.31	1.37
Toluene	0.85	0.92	1.00	1.07	1.15	1.22
Water		0.73	1.00	1.19	1.33	1.50

\* Average temperature of the film = (temperature of the retreating wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

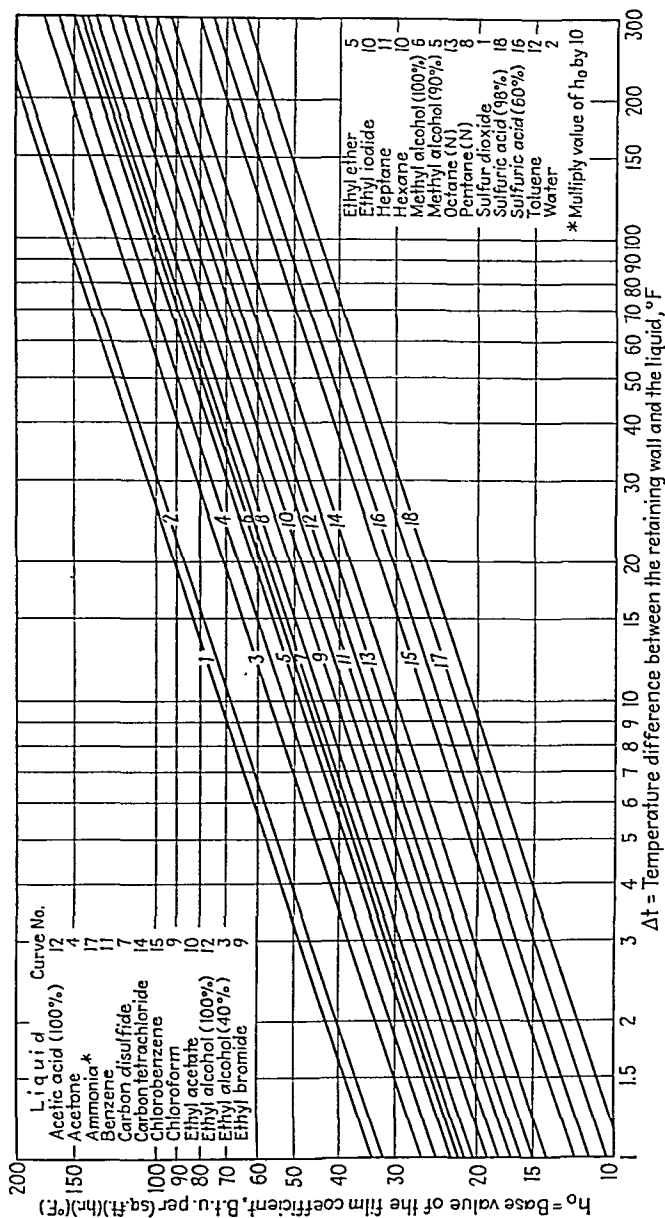


FIG. 33.—Base value of the film coefficient for Case 13.

**Case 14** Gases Heated inside or outside Vertical Tubes or on Vertical Plates, Natural Convection —The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_p$$

where  $h$  = the film coefficient, B t u / (sq ft) (hr) (°F)

$h_0$  = the base value of the film coefficient, from Fig 34

$F_t$  = the temperature-correction factor, from Table 32

$F_p$  = the pressure-correction factor, from Table 33

Note that the film coefficient  $h$  obtained from the foregoing equation is for heat transfer by convection only. If the heat-transfer surface is exposed to other surfaces at different temperatures, it will gain or lose heat by radiation also. The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IV in the Appendix, times the radiation coefficient  $h_r$ , obtained from Fig 10 in Chap II, onto the convection coefficient  $h$  obtained from the equation above.

TABLE 32—TEMPERATURE-CORRECTION FACTOR FOR CASE 14

Gas	Ave temp of the film*						
	-100°F	0°F	100°F	200°F	300°F	400°F	500°F
	$F_t$ †						
Acetone			1.00	1.00	1.13	1.21	1.28
Acetylene	1.03	1.01	1.00	0.93	0.88	0.88	0.87
Air	1.15	1.07	1.00	0.94	0.89	0.85	0.82
Ammonia	1.00	1.00	1.00	0.99	0.98	0.97	0.96
Benzene			1.00	1.10	1.22	1.35	1.48
Butane		0.89	1.00	1.01	1.03	1.04	1.06
Carbon dioxide	1.04	1.02	1.00	0.98	0.95	0.93	0.91
Carbon monoxide	1.14	1.04	1.00	0.94	0.89	0.85	0.81
Chlorine		1.07	1.00	0.95	0.90	0.87	0.84
Chloroform			1.00	1.01	1.02	1.04	1.06
Ethane	0.93	0.96	1.00	1.01	1.07	1.10	1.14
Ethyl acetate			1.00	1.08	1.15	1.22	1.27
Ethyl alcohol			1.00	0.94	0.88	0.82	0.78
Ethyl chloride			1.00	0.99	0.97	0.96	0.94
Ethylene	0.95	0.98	1.00	1.02	1.05	1.07	1.10
Ethyl ether			1.00	1.03	1.08	1.14	1.20
Helium			1.00	0.95	0.90		
Hydrogen	1.18	1.08	1.00	0.93	0.88	0.82	0.78
Hydrogen sulfide		1.05	1.00	0.95	0.89		
Methane	1.05	1.02	1.00	0.97	0.95	0.93	0.91
Methyl chloride		0.91	1.00	1.07	1.13	1.18	1.23
Nitric oxide	1.15	1.07	1.00	0.94	0.88	0.83	0.79
Nitrogen	1.17	1.07	1.00	0.93	0.87	0.82	0.77
Nitrous oxide	1.26	1.11	1.00	0.91	0.84	0.77	
Oxygen	1.15	1.07	1.00	0.94	0.88	0.83	0.79
Pentane (iso)		0.98	1.00	1.04	1.11	1.19	1.28
Steam			1.00	0.96	0.93	0.91	0.89
Sulfur dioxide			1.00	0.96	0.93	0.90	0.88

\* Average temperature of the film = (temperature of the retaining wall + temperature of the main body of the gas) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 33—PRESSURE-CORRECTION FACTOR FOR CASE 14

Pressure lb per sq in gauge	$F_p$	Pressure lb per sq in gauge	$F_p$
0	1.00	100	3.93
10	1.42	150	5.0
20	1.77	200	6.0
30	2.10	250	6.9
40	2.40	300	7.7
50	2.68	350	8.5
60	2.91	400	9.3
70	3.21	450	10.0
80	3.46	500	10.7
90	3.70		

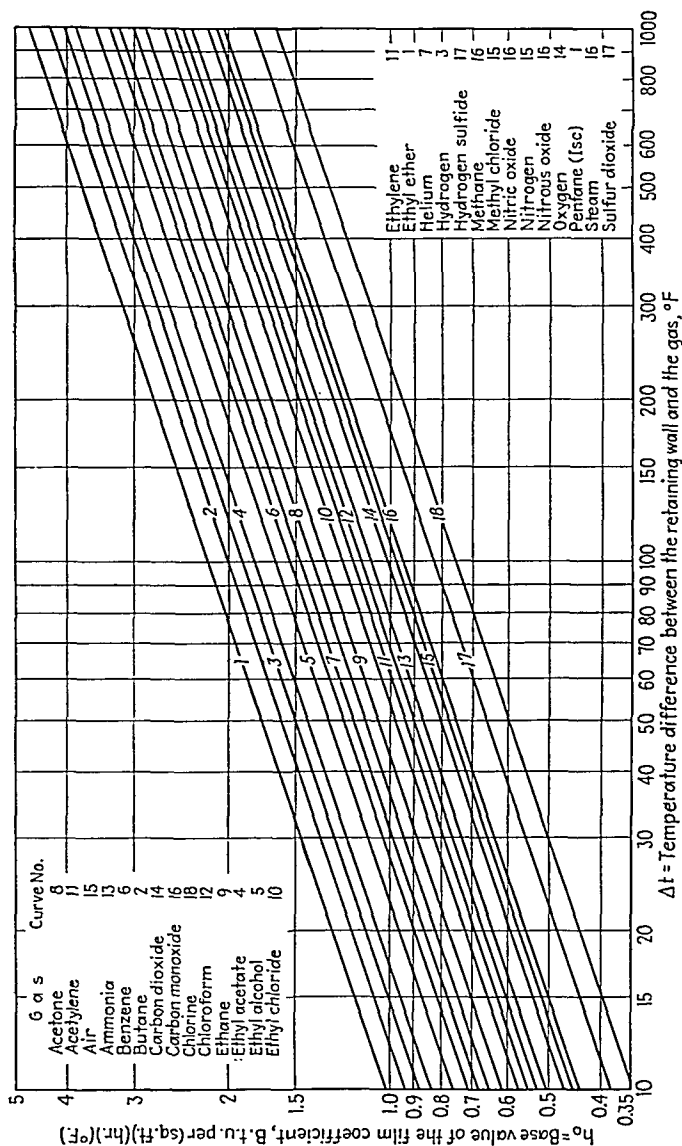


FIG. 34.—Base value of the film coefficient for Case 14.

**Case 15** Liquids Heated or Cooled inside Coils Turbulent Flow — Approximate values of the film coefficient can be determined by the equation

$$h_{\text{coils}} = 1.2 \times h_{\text{ubes}}$$

where  $h_{\text{ubes}}$  for heating is obtained from Case 1 and  $h_{\text{ubes}}$  for cooling is obtained from Case 2

**Case 16** Gases Heated or Cooled inside Coils, Turbulent Flow — The film coefficient can be determined by the equation

$$h_{\text{coils}} = \left[ 1 + 3.54 \left( \frac{d}{d_c} \right) \right] \times h_{\text{ubes}}$$

where  $d$  = the inside diameter of the pipe or tube in

$d_c$  = the diameter of the coil in

$h_{\text{ubes}}$  = the film coefficient for gases inside straight tubes obtained from Case 3 or 4 B t u / (sq ft) (hr) (°F)

**Case 17** Liquids or Gases Heated or Cooled outside Coils, Natural or Forced Convection — The film coefficients for liquids or gases heated or cooled *outside coils* are approximately equal to the film coefficients for liquids or gases heated or cooled *outside single tubes* and can be obtained from Case 5 6 7 11 or 12

**Case 18** Air Heated on Horizontal Plates, Natural Convection — The film coefficient for air heated by natural convection on large flat plates (3 sq ft or more) can be determined by the equations

$$h_H = 1.27 \times h_V \text{ (plate facing upward)}$$

$$\text{or } h_H = 0.67 \times h_V \text{ (plate facing downward)}$$

where  $h_H$  = the film coefficient for horizontal plates B t u / (sq ft) (hr) (°F)

$h_V$  = the film coefficient for vertical plates obtained from Case 14 B t u / (sq ft) (hr) (°F)

Note that the film coefficients obtained from the foregoing equations are for heat transfer by *convection* only. If the heat-transfer surface is exposed to other surfaces at different temperatures it will gain or lose heat by *radiation* also. The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface obtained from Table IV in the Appendix times the radiation coefficient  $h$  obtained from Fig 10 in Chap II onto the convection coefficient  $h$  obtained from the equation above.

**Case 19. Petroleum Oils Heated inside Horizontal or Vertical Tubes, Streamline Flow.**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d,$$

where  $h$  = the film coefficient, based on the arithmetic mean of the terminal temperature differences, B.t.u./ (sq. ft.) (hr.) (°F.);  $h_0$  = the base value of the film coefficient, from Fig. 35;  $F_t$  = the temperature-correction factor, from Table 34;  $F_d$  = the diameter-correction factor, from Table 35.

TABLE 34.—TEMPERATURE-CORRECTION FACTOR FOR CASE 19

Ave. temp. of oil, °F.	$F_t$
100	1.00
200	1.00
300	0.99
400	0.98
500	0.97

TABLE 35.—DIAMETER-CORRECTION FACTOR FOR CASE 19

Inside tube dia., in.	$F_d$	Inside tube dia., in.	$F_d$
0.2	5.00	1.2	0.83
0.3	3.33	1.3	0.77
0.4	2.50	1.4	0.71
0.5	2.00	1.5	0.67
0.6	1.67		
0.7	1.43	2.0	0.50
0.8	1.25	2.5	0.40
0.9	1.11	3.0	0.33
1.0	1.00	3.5	0.27
1.1	0.91	4.0	0.25

TABLE 36.—MAXIMUM PERMISSIBLE VALUES OF  $W'/d$  FOR CASE 19  
 $W'$  = rate of flow per tube, lb. per hr. per tube;  $d$  = inside tube dia., in.

Viscosity, Saybolt Universal seconds	Gravity, °A.P.I.					
	10°	20°	30°	40°	50°	60°
	$\left(\frac{W'}{d}\right)_{\max}$					
40	1,300	1,400	1,500	1,600	1,700	1,800
50	2,300	2,500	2,700	2,800	3,000	3,200
60	3,200	3,500	3,700	3,900	4,100	4,400
80	4,800	5,200	5,500	5,800	6,200	6,500
100	6,400	6,800	7,300	7,800	8,200	8,700
150	10,100	10,900	11,600	12,300	13,000	13,700
200	13,800	14,800	15,700	16,700	17,700	18,700
300	20,600	22,100	23,600	25,000	26,400	27,900
400	27,600	29,600	31,500	33,400	35,400	37,400

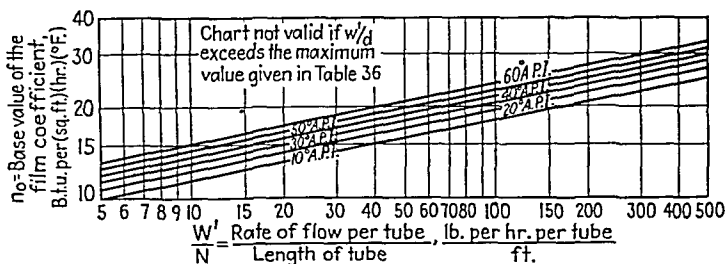


FIG. 35.—Base value of the film coefficient for Case 19.

**Case 20. Petroleum Oils Cooled inside Horizontal or Vertical Tubes, Streamline Flow.**—Approximate values of the film coefficient can be determined by the equation

$$h_c = 0.7 \times h_H,$$

where  $h_c$  = the film coefficient for cooling, B.t.u./ (sq. ft.) (hr.) (°F.);  $h_H$  = the film coefficient for heating, obtained from Case 19, B.t.u./ (sq. ft.) (hr.) (°F.).



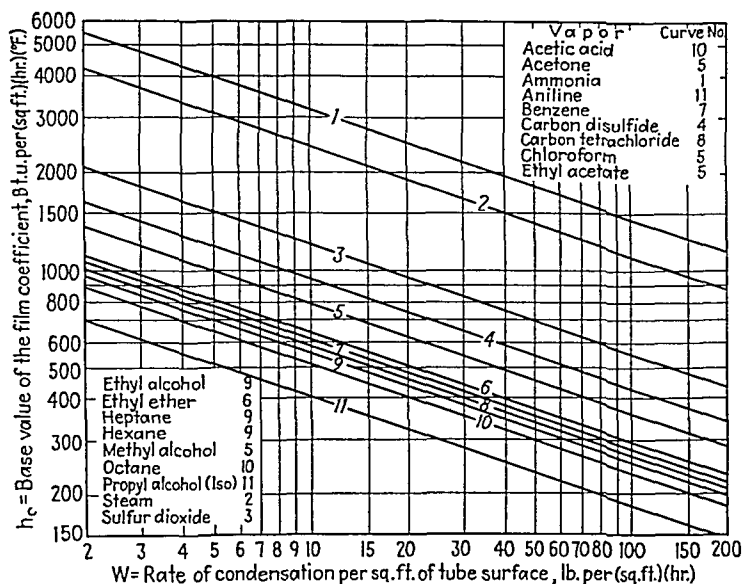


FIG. 36.—Base value of the film coefficient for Cases 21 and 22b.

**Case 22 Condensation of Pure Saturated Vapors on Vertical Tubes or Plates**—The method of evaluating the film coefficient for this case depends upon whether  $W \times N$  is greater or less than the critical value in Table 40

**TABLE 40—CRITICAL VALUE OF THE PRODUCT  $W \times N$  FOR CASE 22**  
 $W$  = rate of condensation per square foot of heat-transfer surface, lb/(sq ft)(hr)  
 $N$  = height of the surface, ft

Vapor	Temp of condensate film*					
	50°F	100°F	150°F	200°F	250°F	300°F
	$(W \times N)_{critical}$					
Acetic acid	1150	770	550	390	270	200
Acetone	290	220	170	140	110	90
Ammonia	80	50	30	20	10	10
Aniline		2300	1100	620	370	220
Benzene	580	410	290	210	160	120
Carbon disulfide	310	260	220	190	170	140
Carbon tetrachloride	900	630	450	310	220	170
Chloroform	500	380	290	230	190	160
Ethyl acetate	420	300	220	170	140	110
Ethyl alcohol	1080	690	430	280	190	130
Ethyl ether	200	160	120	100	80	70
Heptane	360	270	200	160	130	110
Hexane	270	210	170	130	110	90
Methyl alcohol	540	360	260	190	140	110
Octane	500	360	270	210	160	130
Propyl alcohol (iso)		1080	550	340	230	170
Steam	1070	540	340	230	180	150
Sulfur dioxide	280	230	190	150	120	100

\* May be assumed equal to the temperature of the tube wall

a If  $W \times N$  is greater than the critical value given in Table 40

$$h = h_0 \times F_t \times F_N'$$

where  $h$  = the film coefficient Btu/(sq ft)(hr)(°F)

$h_0$  = the base value of the film coefficient, from Fig 37

$F_t$  = the temperature-correction factor, from Table 41

$F_N'$  = the correction factor for the height of the surface from Table 42

**TABLE 41—TEMPERATURE-CORRECTION FACTOR FOR CASE 22a**

Vapor	Temp of condensate film*					
	50°F	100°F	150°F	200°F	250°F	300°F
	$F_t$ †					
Acetic acid		1.00	1.25	1.49	1.81	2.19
Acetone	0.81	1.00	1.29	1.41	1.61	1.82
Ammonia	0.60	1.00	1.56	2.38		
Aniline		1.00	1.50	4.29	7.70	
Benzene	0.69	1.00	1.40	1.88	2.53	3.40
Carbon disulfide	0.87	1.00	1.12	1.23	1.33	1.43
Carbon tetrachloride	0.71	1.00	1.25	1.59		
Chloroform	0.77	1.00	1.29	1.65		
Ethyl acetate	0.71	1.00	1.33	1.71	2.16	2.60
Ethyl alcohol	0.54	1.00	1.58	2.34	3.50	4.95
Ethyl ether	0.79	1.00	1.22	1.46	1.71	1.98
Heptane	0.75	1.00	1.30	1.63	1.98	2.36
Hexane	0.78	1.00	1.24	1.49	1.75	2.00
Methyl alcohol	0.69	1.00	1.44	1.97	2.60	3.30
Octane	0.75	1.00	1.29	1.63	2.00	2.40
Propyl alcohol (iso)		1.00	2.00	3.16	4.45	
Steam	0.46	1.00	1.69	2.62	3.75	4.90
Sulfur dioxide	0.89	1.00	1.07	1.11	1.14	1.16

\* May be assumed equal to the temperature of the wall

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 42.—CORRECTION FACTOR FOR THE HEIGHT OF THE SURFACE,  
CASE 22a

Height of surface, ft.	$F_N'$	Height of surface, ft.	$F_N'$
2	0.53	12	1.08
4	0.69	14	1.15
6	0.82	16	1.21
8	0.91	18	1.27
10	1.00	20	1.32

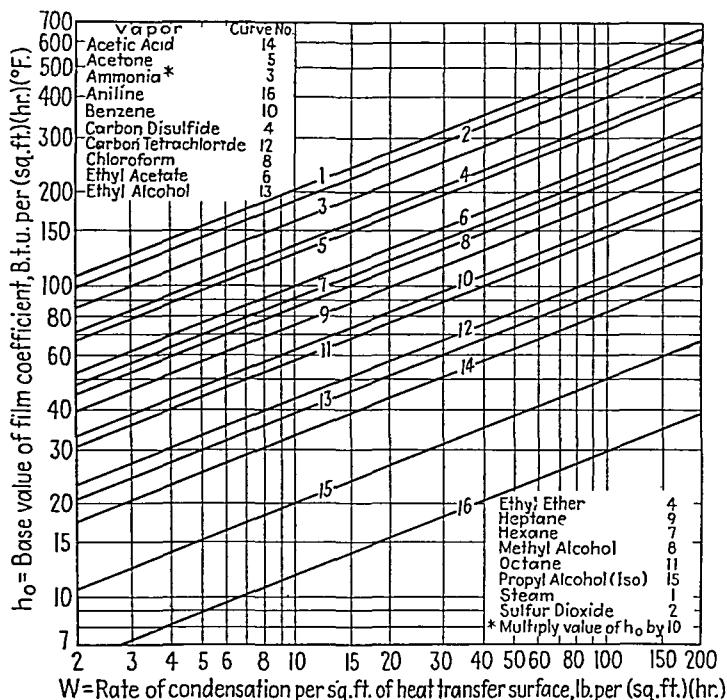


FIG. 37.—Base value of the film coefficient for Case 22a.

b. If  $W \times N$  is less than the critical value given in Table 40,

$$h = 0.29 \times h_0 \times F_t \times F_N'',$$

where  $h$  = the film coefficient, B.t.u./ (sq. ft.) (hr.) (°F.).

$h_0$  = the base value of the film coefficient, from Fig. 36.

$F_t$  = the temperature-correction factor, from Table 37.

$F_N''$  = the correction factor for the height of the surface, from Table 43.

TABLE 43.—CORRECTION FACTOR FOR THE HEIGHT OF THE SURFACE,  
CASE 22b

Height of surface, ft.	$F_N''$	Height of surface, ft.	$F_N''$
2	1.71	12	0.94
4	1.36	14	0.89
6	1.19	16	0.85
8	1.08	18	0.82
10	1.00	20	0.79

**Case 23 Liquids Boiling on Horizontal or Vertical Plates**—Conservative values of the film coefficient for liquids boiling at atmospheric pressure on horizontal or vertical plates can be obtained from Fig 38. At values of  $\Delta t$  below the range given in this figure the film coefficient for boiling is approximately equal to the film coefficient for free convection on vertical surfaces and can be obtained from Case 13.

The film coefficient for water boiling at pressures other than atmospheric can be obtained by multiplying the value obtained from Fig 38 by the pressure-correction factor given in Table 44.

TABLE 44—PRESSURE-CORRECTION FACTOR FOR WATER.  
Absolute Pressure

Atm *	$F_p$
0.2	0.62
0.4	0.78
0.6	0.88
0.8	0.94
1.0	1.00
2.0	1.16
4.0	1.32
6.0	1.40
8.0	1.46
10.0	1.51
15.0	1.60

\* 1 atm = 14.7 lb per sq in.

**Case 24 Liquids Boiling inside Tubes**—If the motion of the liquid is the result of natural convection only, conservative values of the film coefficient can be determined by the equation

$$h_T = 1.25 \times h_P$$

where  $h_T$  = the film coefficient for liquids boiling inside tubes B t u / (sq ft) (hr) (°F)

$h_P$  = the film coefficient for liquids boiling on flat plates obtained from Case 23 B t u / (sq ft) (hr) (°F)

If the liquid is forced through the tubes at high velocities by means of a pump, the film coefficient is approximately the same as it would be if no evaporation took place and can be obtained from Case 1.

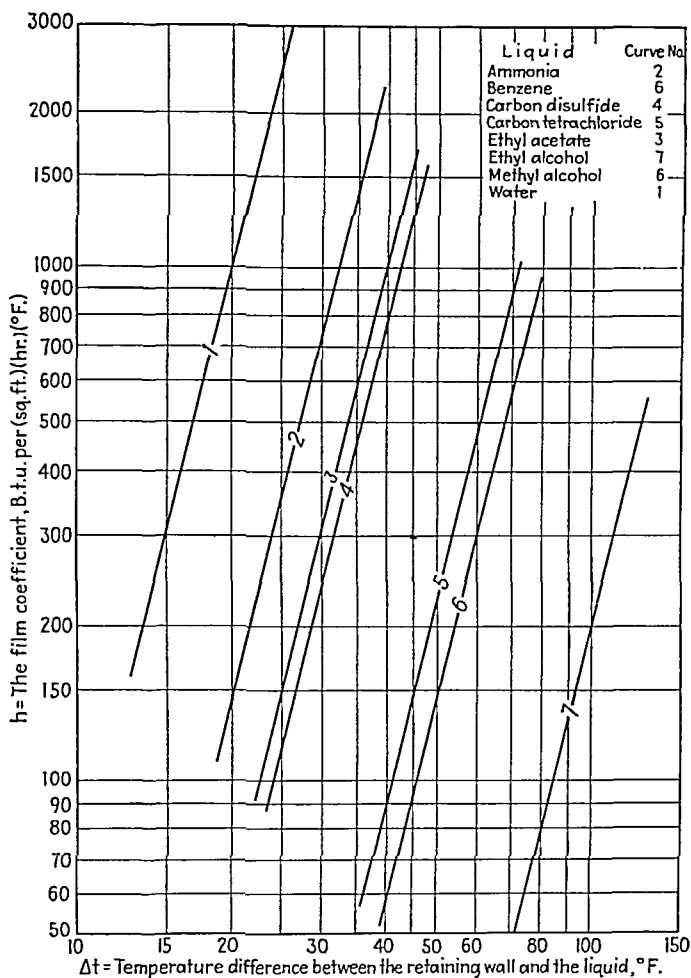


FIG. 38.—Film coefficients for liquids boiling at atmospheric pressure on horizontal or vertical plates, Case 23.

**Case 25 Scale Coefficients**—The scale coefficient  $h_s$  depends upon the thickness and the thermal conductivity of the scale. If the thickness of the scale can be estimated, rough values of the scale coefficient can be calculated by the equation

$$h_s = \frac{k_s}{L_s}$$

where  $h_s$  = the scale coefficient B t u / (sq ft) (hr) (°F)

$k_s$  = the apparent thermal conductivity of the scale, B t u / (ft) (hr) (°F)

$L_s$  = the thickness of the scale, ft

Although the apparent thermal conductivity of scale varies with the kind and with the porosity of the scale, a mean value of 1.0 B t u / (ft) (hr) (°F) may be used for  $k_s$ .

The following values of  $h_s$  may be used for the kinds of convection indicated

Kind of Convection	$h_s$ B t u / (sq ft) (hr) (°F)
Condensing steam	3000
Condensing petroleum vapors	500
Heating muddy water	600
Heating residual petroleum stocks	200

## CHAPTER V

### CALCULATION OF PRESSURE DROP

**33. Introduction.**—High fluid velocities in heat-transfer equipment are desirable because the rate of heat transfer by convection is higher at high than at low velocities. The pressure drop through the equipment is also higher at high velocities, however, and therefore the maximum velocity that can be used is fixed by the permissible pressure drop.

The probable over-all pressure drop through any proposed equipment is equal to the sum of the individual pressure drops that the fluid will undergo. For example, the total pressure drop that the fluid flowing inside the tubes of a heat exchanger will undergo is equal to (1) the pressure drop caused by the abrupt contraction in cross section where the fluid enters the tubes plus (2) the pressure drop caused by friction in the tubes minus (3) the pressure rise caused by the abrupt enlargement in cross section where the fluid leaves the tubes and, if the exchanger is multipass, plus (4) the pressure drop caused by each reversal in the direction of flow. If the fluid is a gas, there will be an additional change in pressure caused by changes in the density of the gas as it is heated or cooled. Thus, changes in density will cause the gas to accelerate or slow down as it flows through the tubes, which will be accompanied by changes in pressure. Charts and tables which can be used to evaluate readily each of these individual pressure drops for various fluids are given at the end of this chapter.

**34. Calculation of the Charts and Tables.**—The equations upon which the charts and tables are based are given in this section. The symbols used in these equations are defined in Table 45.

TABLE 45.—NOMENCLATURE

<i>A</i>	Inside cross-sectional area, sq. ft.
<i>D</i>	Inside tube or pipe diameter, ft.
<i>d</i>	Inside tube or pipe diameter, in.
<i>f</i>	Friction factor, no units.
$\Delta F$	Energy decrease, ft.-lb. per lb.

TABLE 45—NOMENCLATURE—(Continued)

$G$	Weight velocity, lb / (sec) (sq ft of cross section)
$g$	Acceleration of gravity, 32.2 ft per sec <sup>2</sup>
$K$	Contraction coefficient no units
$N$	Length of tube or pipe ft
$P$	Pressure, lb per sq ft abs
$\Delta P$	Pressure rise or drop, lb per sq ft
$R$	Gas constant, ft per °I
$T$	Temperature, °I abs
$t$	Temperature, °F
$t_w$	Tube-wall temperature, °F
$v$	Linear velocity, ft per sec
$v$	Specific volume, cu ft per lb
$W$	Rate of flow, lb per hr per tube
$\mu$	Absolute viscosity lb / (ft) (sec)
$\rho$	Density, lb per cu ft

**CASE 1 LIQUIDS, PRESSURE DROP INSIDE SMOOTH TUBES, TURBULENT FLOW**—For isothermal, turbulent flow of fluids inside straight, smooth tubes, the pressure drop per foot of length can be calculated by the Fanning equation

$$\frac{\Delta P}{N} = \frac{4f\rho V^2}{2gD} \quad (48)$$

The friction factor  $f$  has been determined experimentally by numerous investigators, and for the range  $DV\rho/\mu$  equal to 4,000 to 1,000,000, their results can be represented approximately by the equation

$$f = \frac{0.0653}{\left(\frac{DV\rho}{\mu}\right)^{0.218}} \quad (49)$$

This equation is valid only for isothermal flow, but, as shown by McAdams,<sup>1</sup> it can also be used as an approximation for the heating or cooling of any liquid if the viscosity  $\mu$  is evaluated at the arithmetic average of the tube-wall temperature and the temperature of the main body of the liquid. Figure 39 and Tables 47 and 48 are based on Eqs. (48) and (49), and the values of  $V \times d$  corresponding to  $DV\rho/\mu$  equal to 4,000 are given in Table 49.

**CASE 2 PETROLEUM OILS, PRESSURE DROP INSIDE TUBES OR PIPES, STREAMLINE FLOW**—For streamline flow of petroleum oils inside straight tubes or pipes, McAdams<sup>1</sup> recommends that

<sup>1</sup> McAdams, W. H. "Heat Transmission," p. 112. McGraw-Hill Book Company Inc. 1933.



the pressure drop per foot of length be calculated by the equation

$$\frac{\Delta P}{N} = \frac{35.2\mu' V}{gD^2}, \quad (50)$$

where  $\mu'$  is evaluated at the average temperature  $t'$  defined by the equation

$$t' = \frac{3t + t_r}{4}. \quad (51)$$

In the latter equation,  $t$  represents the temperature of the main body of the oil and  $t_r$  represents the temperature of the tube wall. Equation (50) is valid only if the value of  $DV\rho/\mu$  is less than 2,100. Figure 40 and Table 50 are based on Eq. (50) and on the equation

$$W' = \frac{\pi D^2}{4} \times V \times 3,600\rho. \quad (52)$$

The values of  $W'/d$  corresponding to  $DV\rho/\mu$  equal to 2,100 are given in Table 51.

CASE 3. LIQUIDS, PRESSURE DROP AT THE ENTRANCE TO TUBES.—The pressure drop caused by an abrupt contraction in cross section, such as occurs at the entrance to unflared tubes, can be calculated by the equation

$$\Delta P = \left[ \frac{(V_2^2 - V_1^2)}{2g} + K \left( \frac{V_2^2}{2g} \right) \right] \times \rho, \quad (53)$$

where  $V_1$  is the average velocity in the larger section and  $V_2$  is the average velocity in the smaller. The first term in the

TABLE 46.—CONTRACTION COEFFICIENT FOR SHARP-EDGED ENTRANCES\*

Ratio of Larger to Smaller Area	$K$
1 0	0 00
1 5	0 13
2 0	0 19
2 5	0 24
3 0	0.27
3 5	0 30
4.0	0.32
6.0	0.38
8 0	0.41
10 0	0 43

\* Taken from H. J. Hughes and A. T. Safford, "Hydraulics," The Macmillan Company 1926.

right-hand member of this equation represents the pressure drop required to increase the velocity of the fluid, and the second represents the pressure drop caused by friction. Values of the contraction coefficient  $K$  for sharp-edged entrances are given in Table 46. Figure 41 and Table 52 are based on Eq. (53), on these values of  $K$ , and on the equation

$$A_1 V_1 = A_2 V_2 \quad (54)$$

For rounded or conical entrances, the value of  $K$  is quite small, and the pressure drop caused by friction is practically negligible.

**CASE 4 LIQUIDS, PRESSURE RISE AT THE OUTLET FROM TUBES**—The rise in pressure caused by an abrupt enlargement in cross section, such as occurs at the outlet from tubes, can be calculated by the equation

$$\Delta P = \left[ \frac{(V_1^2 - V_2^2)}{2g} - \frac{(V_1 - V_2)^2}{2g} \right] \times \rho, \quad (55)$$

where  $V_1$  is the average velocity in the smaller section and  $V_2$  is the average velocity in the larger. The first term in the right-hand member of this equation represents the increase in pressure caused by the decrease in the velocity of the fluid, and the second represents the pressure drop caused by friction. It may be noted that the first term is always larger than the second and that  $\Delta P$  therefore always represents a pressure rise. Figure 42 and Table 53 are based on Eqs. (54) and (55).

**CASE 5 GASES, PRESSURE DROP INSIDE SMOOTH TUBES, TURBULENT FLOW**—Figure 43 and Tables 54 and 55 are based on Eqs. (48) and (49) and on the assumption that the gases obey the perfect-gas law  $Pv = RT$ . Although Eq. (49) is based on data for isothermal flow, it may also be used if the gases are being heated or cooled, because the viscosities of gases do not change rapidly with temperature.

**CASE 6 GASES, PRESSURE DROP AT THE ENTRANCE TO TUBES**—Figure 44 and Table 56 are also based on Eqs. (53) and (54) and on the values of the contraction coefficient  $K$  given in Table 46. The use of Eq. (54) involves the assumption that the density of the gas changes only slightly during the contraction. This assumption is permissible for moderate rates of flow. The values of the constant  $b$  given in Table 57 are based on the assumption that the gases obey the perfect-gas law  $Pv = RT$ .

CASE 7. GASES, PRESSURE RISE AT THE OUTLET FROM TUBES.—Figure 45 and Table 58 are also based on Eqs. (54) and (55). In using Eq. (54), it is again assumed that the density of the gas changes only slightly during the process.

CASE 8. GASES, PRESSURE RISE OR DROP CAUSED BY CHANGES IN DENSITY.—If a gas is heated as it flows through a tube of constant cross section, it will leave the tube at a higher velocity than it enters, because its density will be decreased. This increase in velocity will be accompanied by a drop in pressure which can be determined as follows: The energy required to accelerate the gas is equal to the increase in the kinetic energy of the gas; *i.e.*,

$$\Delta F = \frac{V_2^2}{2g} - \frac{V_1^2}{2g},$$

where  $\Delta F$  is measured in foot-pounds per pound of gas. Since the weight rate of flow  $G = V_1/v_1 = V_2/v_2$ , the foregoing equation can be written

$$\begin{aligned}\Delta F &= \frac{G^2}{g} (v_2 - v_1) \left( \frac{v_2 + v_1}{2} \right) \\ &= \frac{G^2}{g} (v_2 - v_1) \times v_{ave},\end{aligned}$$

or

$$\Delta P = \frac{G^2}{g} (v_2 - v_1). \quad (56)$$

Equation (56) also applies if the gas is cooled, the negative value obtained indicating that the pressure rises.

Figure 46 is based on Eq. (56). This equation can also be applied to liquids; but since the densities of liquids vary only slightly with changes in temperature, the values of  $\Delta P$  are negligible.

CASE 9. PRESSURE DROP CAUSED BY A REVERSAL IN THE DIRECTION OF FLOW.—The pressure drop caused by a reversal in the direction of flow depends upon the design of the passage in which the reversal takes place. This loss may be relatively large for liquids but is usually negligible for gases.

CASE 10. LIQUIDS OR GASES, PRESSURE DROP IN ANNULAR SPACES OR OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW PARALLEL TO TUBES.—For *turbulent* flow, the pressure drop in annular spaces or outside tube bundles can be determined from

Note that this value does not include the loss of head due to differences in the elevation of the first and last headers

**Illustrative Problem 2**—A two-stage air compressor handles 500 cu ft of free air per minute (measured at 70°F and atmospheric pressure). The air is cooled between the stages from 240 to 90°F in an intercooler having the following specifications

Total number of tubes = 40

Length of tubes = 8 ft

Diameter of tubes =  $\frac{5}{8}$  in o d

Tube wall thickness = No. 16 B W G

Ratio of cross sectional area of headers

to cross sectional area of tubes = 2.5

Determine the pressure drop through the intercooler if the average pressure of the air in the intercooler is 25 lb per sq in gauge

**Solution**—The density of the air entering the compressor (see Table 57) is  $2.70 \times 14.7 / (70 + 460) = 0.075$  lb per cu ft, and the cross-sectional area of each tube is 0.1924 sq in. Therefore, the weight velocity in the tubes is

$$G = \frac{500 \times 0.075}{60} \times \frac{144}{40 \times 0.1924} \\ = 11.7 \text{ lb / (sec) (sq ft)}$$

The density of the air entering the intercooler is

$$2.70 \times \frac{(25 + 14.7)}{(240 + 460)} = 0.153 \text{ lb per cu ft,}$$

and the density of the air leaving is  $2.70 \times (25 + 14.7) / (90 + 460) = 0.202$  lb per cu ft. The average density of the air in the intercooler is 0.178 lb per cu ft.

The pressure drop caused by friction inside the tubes can be determined by Case 5. Thus from Fig. 43  $(\Delta p / L)_s = 0.048$  lb (sq in) / (ft), and from Tables 54 and 55  $F_r = 1.14$  (average air temperature = 165°F) and  $F_d = 2.38$  (inside tube diameter = 0.495 in). The absolute pressure of the air is  $(25 + 14.7) / 14.7 = 2.70$  atm abs. Therefore, since the tubes are 8 ft long

$$\Delta p_1 = 0.048 \times 1.14 \times 2.38 \times \frac{1}{2.70} \times 8 \\ = 0.39 \text{ lb per sq in}$$

The pressure drop at the entrance to the tubes can be determined by Case 6. Thus, from Fig. 44  $\Delta p_s = 0.12$ , and from Table 56,  $F_r = 0.86$ . Therefore,

$$\Delta p_1 = 0.12 \times 0.86 \\ = 0.10 \text{ lb per sq in}$$

The pressure rise at the outlet from the tubes can be determined by

Case 7. Thus, from Fig. 45,  $\Delta p_0 = 0.035$  lb. per sq. in.; and from Table 58,  $F_r = 1.28$ . Therefore,

$$\begin{aligned}\Delta p_3 &= -0.035 \times 1.28 \\ &= -0.04 \text{ lb. per sq. in.}\end{aligned}$$

The change in pressure caused by the increase in the density of the air can be determined by Case 8. Thus, from Fig. 46,  $p_0' = 0.19$  lb. per sq. in. and  $p_0'' = 0.14$  lb. per sq. in. Therefore,

$$\begin{aligned}\Delta p_4 &= 0.14 - 0.19 \\ &= -0.05 \text{ lb. per sq. in.}\end{aligned}$$

Finally, the total pressure drop through the intercooler is the sum of these individual drops:

$$\begin{aligned}\Delta p &= 0.39 + 0.10 - 0.04 - 0.05 \\ &= 0.40 \text{ lb. per sq. in.}\end{aligned}$$

**Illustrative Problem 3.**—A petroleum oil flows through the tubes of a heater at the rate of 1,500 lb. per hr. per tube. The tubes are 10 ft. long,  $\frac{3}{4}$  in. o.d., have No. 18 B.W.G. walls (0.652 in. i.d.), and are at an average temperature of 200°F. Estimate the pressure drop through the tubes if the oil is heated from 60 to 160°F. and has the following properties:

Gravity = 20° A.P.I.

Viscosity at 95°F. = 200 Saybolt Universal seconds.

Viscosity at 160°F. = 60 Saybolt Universal seconds.

Viscosity at 170°F. = 55 Saybolt Universal seconds.

*Solution.*—The pressure drop can be determined by Case 2 since  $(W'/d)$  is less than the maximum permissible value given in Table 51. Thus, at 160°F.,  $(W'/d)_{\max} = 3,500$ , while  $(W'/d) = 1,500/0.652 = 2,300$ .

The "film" temperature of the oil on entering the tubes is

$$\frac{(3 \times 60 + 200)}{4} = 95^\circ\text{F.},$$

and the viscosity at this temperature is 200 Saybolt Universal seconds. Hence, from Fig. 40,  $(\Delta p/N)_0 = 0.037$  lb./((sq. in.)/(ft.)); and from Table 50,  $F_d = 5.50$ . Therefore,  $(\Delta p/N) = 0.037 \times 5.50 = 0.204$  lb./((sq. in.)/(ft.)).

Similarly, the film temperature of the oil on leaving the tubes is

$$\frac{(3 \times 160 + 200)}{4} = 170^\circ\text{F.},$$

the viscosity is 55 Saybolt Universal seconds,  $(\Delta p/N)_0 = 0.0075$  lb./((sq. in.)/(ft.)), and  $(\Delta p/N) = 0.0075 \times 5.50 = 0.041$  lb./((sq. in.)/(ft.)).

The average pressure drop per foot of length is somewhere between 0.041 and 0.204 lb./((sq. in.)/(ft.)). Taking the arithmetic average of these two values, the total pressure drop is probably close to  $10 \times 0.123 = 1.23$  lb. per sq. in.

**Case 1. Liquids, Pressure Drop inside Smooth Tubes, Turbulent Flow.**—The pressure drop per foot of length can be determined by the equation  $(\Delta p/N) = (\Delta p/N)_0 \times F_t \times F_d$ , where  $(\Delta p/N)$  = the pressure drop per foot of length, lb/(sq in)(ft of length),  $(\Delta p/N)_0$  = the base value of the pressure drop per foot of length, from Fig 39,  $F_t$  = the temperature-correction factor, from Table 47,  $F_d$  = the diameter-correction factor, from Table 48

TABLE 47—TEMPERATURE-CORRECTION FACTOR FOR CASE 1

Liquid	Density at 70°F lb per cu ft	Ave temp of the film*					
		0°F	50°F	100°F	150°F	200°F	250°F
		<i>F<sub>t</sub></i>					
Acetic acid (100%)	65.4			1.00	0.90	0.82	0.76
Acetic acid (50%)	66.0		1.20	1.00	0.87	0.77	0.69
Acetone	49.2	1.18	1.09	1.00	0.90	0.79	0.67
Ammonia	38.6	1.40	1.19	1.00	0.85	0.67	0.52
Amyl acetate	51.0	1.24	1.11	1.00	0.90	0.81	0.72
Amyl alcohol (iso)	50.7	1.60	1.21	1.00	0.85	0.74	0.65
Aniline	64.0		1.17	1.00	0.86	0.74	0.64
Benzene	54.8		1.12	1.00	0.83	0.81	0.76
Brine (CaCl <sub>2</sub> ) (20%)	76.7	1.31	1.13	1.00	0.89	0.79	0.69
Butyl alcohol (n)	19.8	1.45	1.19	1.00	0.87	0.78	0.73
Carbon disulfide	78.7	1.14	1.07	1.00	0.94	0.88	0.82
Carbon tetrachloride	93.5	1.32	1.14	1.00	0.90	0.84	0.81
Chlorobenzene	73.0	1.27	1.13	1.00	0.89	0.81	0.76
Chloroform	92.7	1.21	1.10	1.00	0.91	0.83	0.76
Ethyl acetate	56.3	1.25	1.11	1.00	0.91	0.84	0.79
Ethyl alcohol (100%)	49.2	1.34	1.15	1.00	0.87	0.77	0.71
Ethyl alcohol (40%)	60.8	1.45	1.22	1.00	0.78	0.67	
Ethyl bromide	90.8	1.17	1.08	1.00	0.92	0.84	0.77
Ethylene glycol (50%)	66.5	1.49	1.22	1.00	0.82	0.69	0.61
Ethyl ether	44.4	1.22	1.10	1.00	0.92	0.85	0.79
Ethyl iodide	129.8	1.20	1.09	1.00	0.92	0.85	0.79
Glycerol (50%)	70.3	1.61	1.23	1.00	0.82	0.67	0.53
Heptane	42.6	1.23	1.10	1.00	0.91	0.83	0.75
Hexane	41.1	1.19	1.09	1.00	0.91	0.83	0.75
Methyl alcohol (100%)	49.4	1.24	1.12	1.00	0.88	0.76	0.64
Methyl alcohol (90%)	51.1	1.25	1.12	1.00	0.89	0.80	0.73
Methyl alcohol (40%)	58.2	1.06	1.23	1.00	0.85	0.74	0.64
Octane (n)	43.8	1.18	1.09	1.00	0.91	0.83	0.75
Pentane (n)	39.2	1.27	1.12	1.00	0.91	0.85	0.80
Propyl alcohol (iso)	48.9	1.61	1.25	1.00	0.84	0.74	0.69
Sulfur dioxide	86.0	1.21	1.10	1.00	0.90	0.81	0.78
Sulfuric acid (60%)	93.5		1.18	1.00	0.89	0.81	0.82
Toluene	53.9	1.20	1.10	1.00	0.92	0.85	0.79
Water	62.3		1.17	1.00	0.89	0.81	0.74

\* Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 48—DIAMETER-CORRECTION FACTOR FOR CASE 1

Inside tube dia in	<i>F<sub>d</sub></i>	Inside tube dia in	<i>F<sub>d</sub></i>
0.2	7.20	1.2	0.80
0.3	4.37	1.3	0.72
0.4	3.08	1.4	0.66
0.5	2.34	1.5	0.61
0.6	1.88		
0.7	1.55	2.0	0.43
0.8	1.31	2.5	0.32
0.9	1.14	3.0	0.25
1.0	1.00	3.5	0.22
1.1	0.89	4.0	0.18

TABLE 49—MINIMUM PERMISSIBLE VALUE OF  $V \times d$  FOR CASE 1

$V$  = velocity, ft/sec  $d$  = inside tube dia, in

Liquid*	Ave temp of liquid				
	50°F	100°F	150°F	200°F	250°F
	$(V \times d)_{min}$				
Acetic acid (50%)	1.4	0.7	0.4	0.3	0.2
Amyl alcohol (iso)	3.8	1.6	0.8	0.5	0.3
Aniline	3.0	1.5	0.8	0.4	0.3
Butyl alcohol (n)	2.4	1.3	0.7	0.4	0.2
Ethyl alcohol (100%)	1.0	0.6	0.4	0.3	0.2
Ethyl alcohol (40%)	2.4	0.9	0.5	0.4	0.3
Ethylene glycol (50%)	2.5	1.0	0.5	0.3	0.2
Glycerol (50%)	3.9	1.7	0.9	0.5	0.4
Methyl alcohol (40%)	1.4	0.6	0.3	0.2	0.2
Propyl alcohol (iso)	2.2	0.9	0.5	0.4	0.3
Sulfuric acid (60%)	2.6	1.4	0.9	0.7	0.7

\* Only those liquids for which  $(V \times d)_{min}$  at 100°F is greater than 0.5 are listed

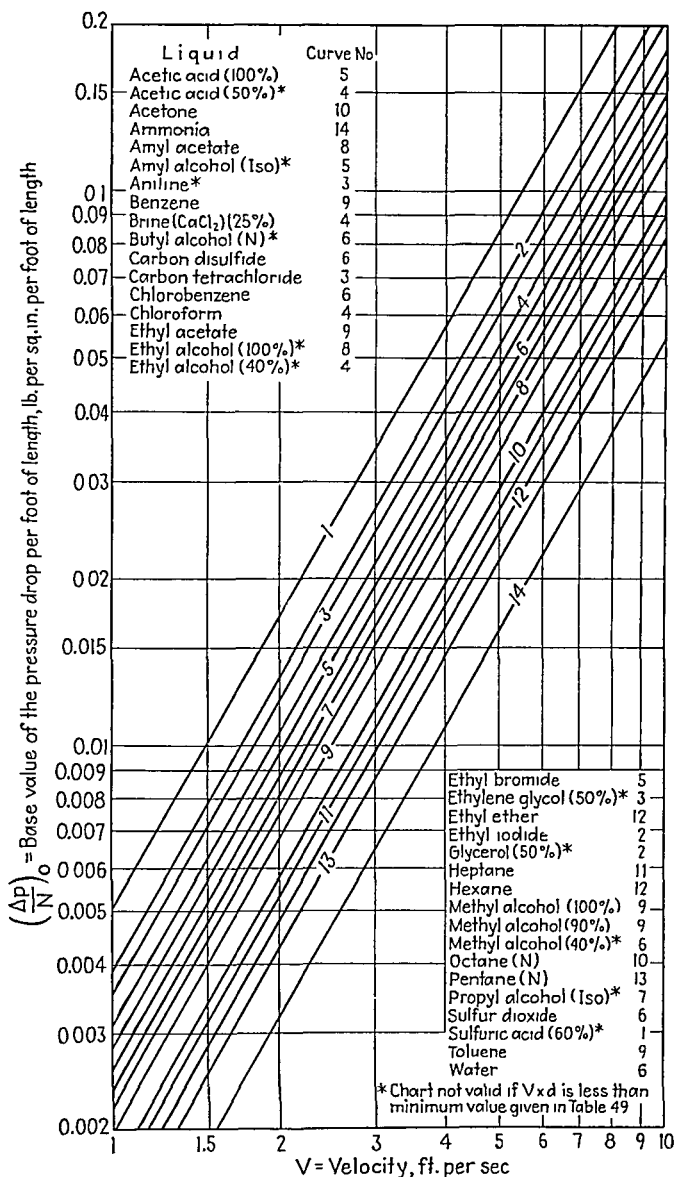


FIG. 39.—Base value of the pressure drop for Case 1.

**Case 2. Petroleum Oils, Pressure Drop inside Tubes or Pipes, Stream-line Flow**—The pressure drop per foot of length can be determined by the equation

$$\left(\frac{\Delta p}{\lambda}\right) = \left(\frac{\Delta p}{\lambda}\right)_0 \times F_d$$

where  $\left(\frac{\Delta p}{\lambda}\right)$  = the pressure drop per foot of length, lb/(sq in.)(ft. of length)

$\left(\frac{\Delta p}{\lambda}\right)_0$  = the base value of the pressure drop per foot of length, from

Fig. 40

$F_d$  = the diameter-correction factor, from Table 50

TABLE 50—DIAMETER-CORRECTION FACTOR FOR CASE 2

Inside Tube Dia., in.	$F_d^*$
0.5	16.00
0.6	7.73
0.7	4.19
0.8	2.44
0.9	1.52
1.0	1.00
1.1	0.64
1.2	0.48
1.3	0.35
1.4	0.28
1.5	0.20

\* If greater accuracy is required than can be obtained by interpolation,  $F_d$  may be calculated by the equation

$$F_d = \frac{1}{d^4}$$

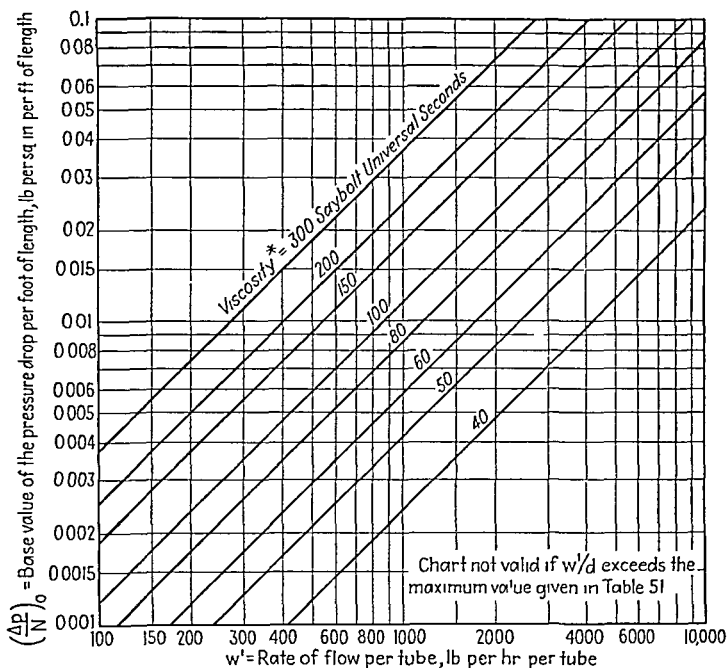
where  $d$  = the inside tube diameter in.

TABLE 51—MAXIMUM PERMISSIBLE VALUES OF  $W/d$  FOR CASE 2

$W$  = rate of flow per tube lb per hr per tube  $d$  = inside tube diameter in.

Viscosity, Saybolt Universal seconds	Gravity *A.P.L.					
	10°	20°	30°	40°	50°	60°
	$\left(\frac{W}{d}\right)_{\max}$					
40	1 300	1 400	1 500	1 600	1 700	1 800
50	2 300	2 500	2 700	2 800	3 000	3 200
60	3 200	3 500	3 700	3 900	4 100	4 400
80	4 800	5 200	5 500	5 800	6 200	6 500
100	6 400	6 800	7 200	7 600	8 000	8 400
150	10 100	10 600	11 100	11 600	12 100	12 600
200	13 800	14 400	15 000	15 600	16 200	16 800
300	20 600	21 400	22 200	23 000	23 800	24 600
400	27 600	28 600	29 600	30 600	31 600	32 600





\* The viscosity should be calculated at an average temperature equal to  $(3t + t_w)/4$ , where  $t$  is the temperature of the oil and  $t_w$  is the temperature of the tube wall

FIG. 40.—Base value of the pressure drop for Case 2

**Case 3 Liquids, Pressure Drop at the Entrance to Tubes.**—The pressure drop at the entrance to unflared tubes (or at any other abrupt contraction in cross section) can be determined by the equation

$$\Delta p = \Delta p_0 \times F_r,$$

where  $\Delta p$  = the pressure drop, lb per sq in

$\Delta p_0$  = the base value of the pressure drop, from Fig 41

$F_r$  = the area-ratio-correction factor, from Table 52

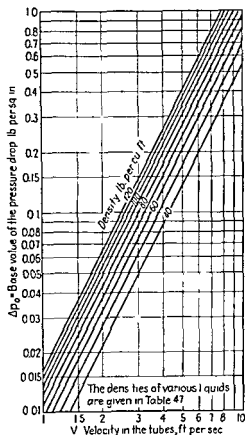


TABLE 52—AREA-RATIO-CORRECTION FACTOR FOR CASE 3

$A_1$  = cross-sectional area before entering the tubes

$A_2$  = cross-sectional area of the tubes.

$\frac{A_1}{A_2}$	$F_r$
1.0	0.00
1.5	0.54
2.0	0.75
2.5	0.86
3.0	0.92
3.5	0.97
4.0	1.00
6.0	1.07
8.0	1.11
10.0	1.13

FIG 41 —Base value of the pressure drop for Case 3

**Case 4. Liquids, Pressure Rise at the Outlet from Tubes.**—The rise in pressure at the outlet from tubes (or at any other abrupt enlargement in cross section) can be determined by the equation

$$\Delta p = \Delta p_0 \times F_r,$$

where  $\Delta p$  = the rise in pressure, lb. per sq. in.

$\Delta p_0$  = the base value of the rise in pressure, from Fig. 42.

$F_r$  = the area-ratio-correction factor, from Table 53.

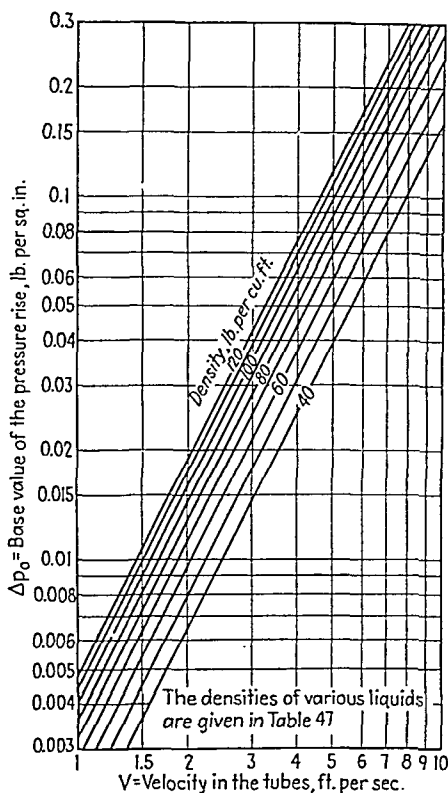


TABLE 53.—AREA-RATIO-CORRECTION FACTOR FOR CASE 4

$A_1$  = cross-sectional area of the tubes.

$A_2$  = cross-sectional area after leaving the tubes.

$\frac{A_2}{A_1}$	$F_r$
1.0	0.00
1.5	1.18
2.0	1.33
2.5	1.28
3.0	1.18
3.5	1.09
4.0	1.00
6.0	0.74
8.0	0.58
10.0	0.48

FIG. 42.—Base value of the pressure rise for Case 4.

**Case 5** Gases, Pressure Drop inside Smooth Tubes, Turbulent Flow —  
The pressure drop per foot of length can be determined by the equation

$$\left(\frac{\Delta p}{N}\right) = \left(\frac{\Delta p}{N}\right)_0 \times F_t \times F_d \times \frac{1}{P}$$

where  $\left(\frac{\Delta p}{N}\right)$  = the pressure drop per foot of length, lb / (sq in) (ft of length)

$\left(\frac{\Delta p}{N}\right)_0$  = the base value of the pressure drop per foot of length, from  
Fig 43

$F_t$  = the temperature-correction factor, from Table 54

$F_d$  = the diameter-correction factor, from Table 55

$P$  = the average pressure of the gas atm abs (1 atm = 14.7 lb per sq in)

TABLE 54 — TEMPERATURE-CORRECTION FACTOR FOR CASE 5

Ave Temp of Gas °F	$F_t$
100	0.60
0	0.80
100	1.00
200	1.21
300	1.42
400	1.64
500	1.86

TABLE 55 — DIAMETER CORRECTION FACTOR FOR CASE 5

Inside Tube Dia In	$F_d$
0.2	7.20
0.3	4.37
0.4	3.08
0.5	2.34
0.6	1.88
0.7	1.55
0.8	1.31
0.9	1.14
1.0	1.00
1.1	0.89
1.2	0.80
1.3	0.72
1.4	0.66
1.5	0.61
2.0	0.43
2.5	0.32
3.0	0.25
3.5	0.22
4.0	0.18

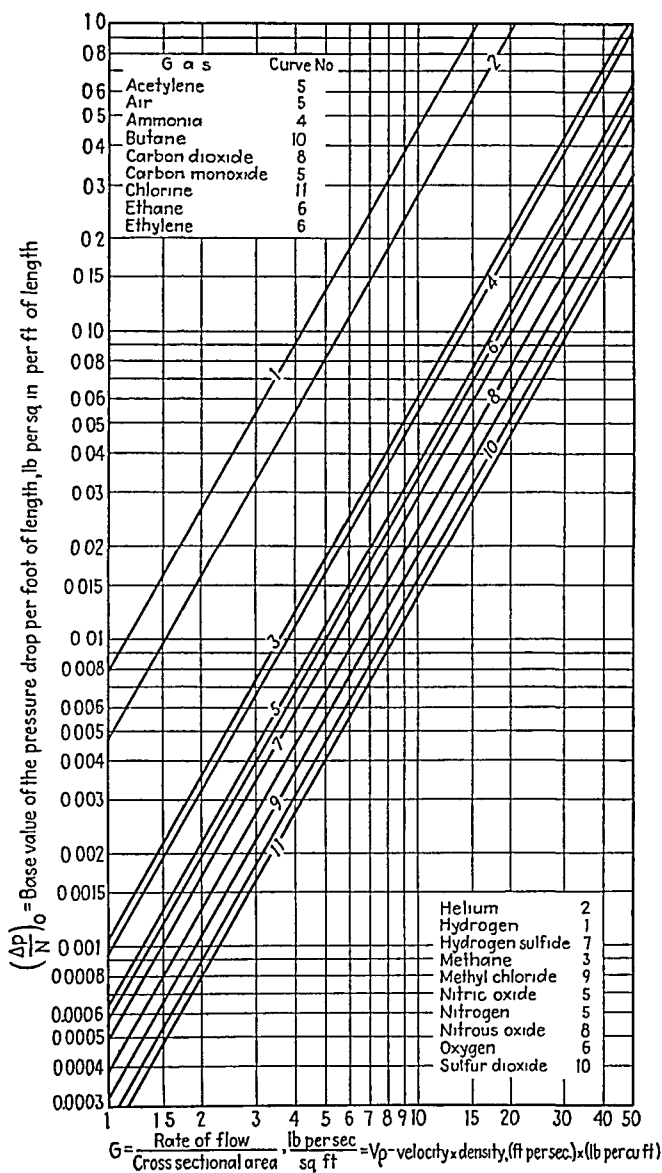


FIG 43 — Base value of the pressure drop for Case 5.

**Case 6. Gases, Pressure Drop at the Entrance to Tubes.**—The pressure drop at the entrance to unflared tubes (or at any other abrupt contraction in cross section) can be determined by the equation

$$\Delta p = \Delta p_0 \times F_r$$

where  $\Delta p$  = the pressure drop lb per sq in

$\Delta p_0$  = the base value of the pressure drop from Fig. 44

$F_r$  = the area ratio-correction factor from Table 56

TABLE 56—AREA RATIO-CORRECTION FACTOR FOR CASE 6

$A_1$  = cross-sectional area before entering the tubes

$A_2$  = cross-sectional area of the tubes

$\frac{A_1}{A_2}$	$F_r$
1.0	0.00
1.5	0.54
2.0	0.75
2.5	0.86
3.0	0.92
3.5	0.97
4.0	1.00
6.0	1.07
8.0	1.11
10.0	1.13

TABLE 57—GAS DENSITIES

Approximate values of the density can be calculated by the equation

$$\rho = \frac{bp}{t + 460}$$

where  $\rho$  = the density of the gas lb per cu ft

$p$  = the absolute pressure of the gas lb/sq in abs

$t$  = the temperature of the gas °F

$b$  = a constant obtained from the following table

Gas	$b$
Acetylene	2.46
Air	2.70
Ammonia	1.61
Butane	5.57
Carbon dioxide	4.13
Carbon monoxide	2.61
Chlorine	6.72
Ethane	2.83
Ethylene	2.63
Helium	0.37
Hydrogen	0.19
Hydrogen sulfide	3.21
Methane	1.50
Methyl chloride	4.82
Nitric oxide	2.80
Nitrogen	2.61
Nitrous oxide	4.13
Oxygen	2.98
Sulfur dioxide	6.11

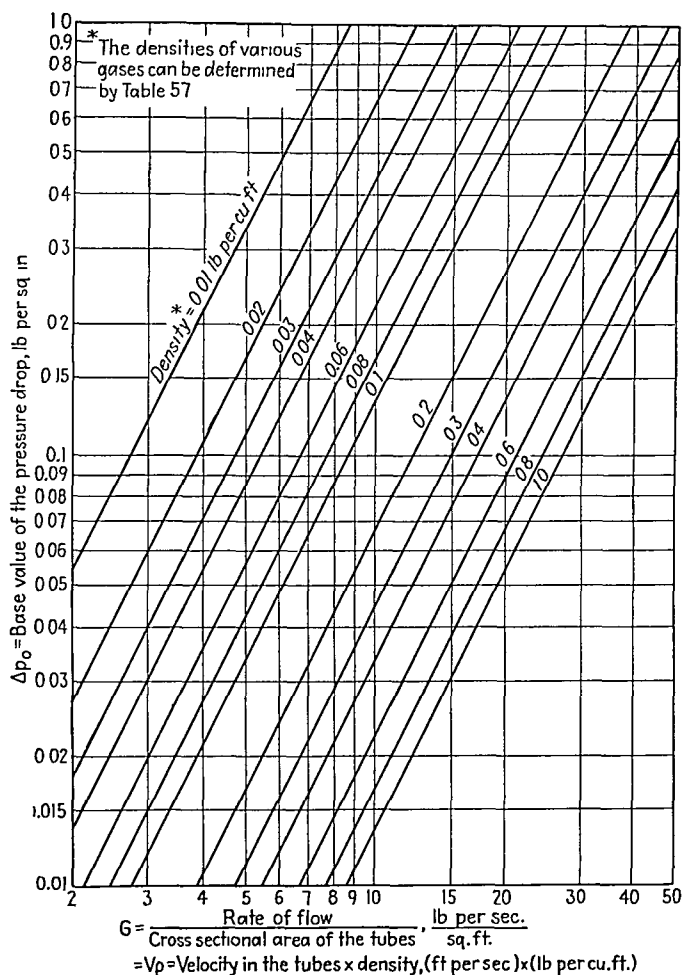


FIG. 44.—Base value of the pressure drop for Case 6.

**Case 7. Gases, Pressure Rise at the Outlet from Tubes.**—The rise in pressure at the outlet from tubes (or at any other abrupt enlargement in cross section) can be determined by the equation

$$\Delta p = \Delta p_s \times F_r$$

where  $\Delta p$  = the rise in pressure, lb per sq in

$\Delta p_s$  = the base value of the rise in pressure, from Fig. 45

$F_r$  = the area ratio-correction factor, from Table 58

TABLE 58—AREA RATIO-CORRECTION FACTOR FOR CASE 7

$A_1$  = cross-sectional area of the tubes

$A_2$  = cross sectional area after leaving the tubes

$\frac{A_2}{A_1}$	$F_r$
1.0	0.00
1.5	1.18
2.0	1.33
2.5	1.28
3.0	1.18
3.5	1.00
4.0	1.00
6.0	0.74
8.0	0.58
10.0	0.48



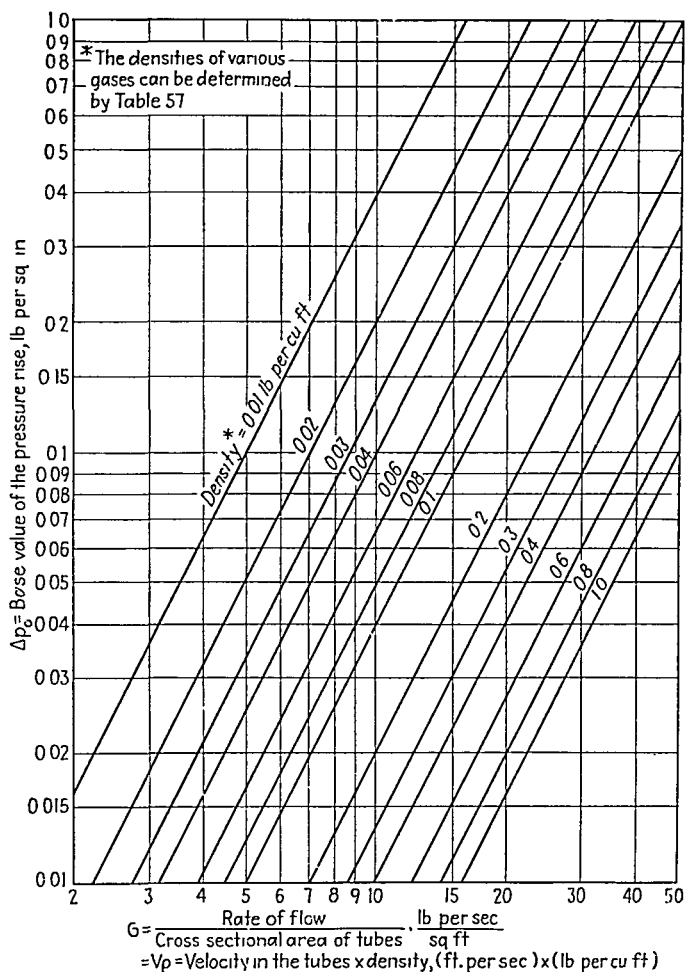


FIG. 45 — Base value of the pressure drop for Case 7

**Case 8 Gases, Pressure Rise or Drop Caused by Changes in Density —**

The change in pressure caused by changes in the density of a gas flowing inside tubes can be determined by the equation

$$\Delta p = p_o' - p_o'',$$

where  $\Delta p$  = the drop in pressure if positive and the rise in pressure if negative, lb per sq in

$p_o'$  = the pressure term for the gas entering the tubes, from Fig. 46

$p_o''$  = the pressure term for the gas leaving the tubes, from Fig. 46

Note that  $\Delta p$  is positive if the gas is heated and negative if cooled

**Case 9 Pressure Drop Caused by a Reversal in the Direction of Flow —**

The pressure drop caused by a reversal in the direction of flow depends upon the design of the passage in which the reversal takes place. If the passage is well rounded the pressure drop is usually negligible, but if the passage has sharp corners, as in the headers of a multipass heat exchanger, the pressure drop may be relatively large.

For design purposes the energy loss in passages with sharp corners may be assumed approximately equal to one-half the kinetic energy of the fluid in the tubes. Based on this assumption, the pressure drop can be estimated by the equation

$$\Delta p = 0.000051 V^2 \rho, \quad (58)$$

where  $\Delta p$  = the pressure drop lb per sq in

$V$  = the velocity in the tubes, ft per sec

$\rho$  = the density of the fluid lb per cu ft

**Case 10 Liquids or Gases, Pressure Drop in Annular Spaces or outside Tube Bundles, Direction of Flow Parallel to Tubes —**The pressure drop per foot of length can be determined from Case 1 or 5 for fluids *inside* tubes if an equivalent diameter  $d_e$  is used in determining the diameter correction factor  $F_d$ . The correct equivalent diameter for flow outside tube bundles can be determined either by Eq. (46) or by Fig. 29 or 30 given in Case 9 in the preceding chapter. The correct equivalent diameter for annular spaces is equal to the difference between the outside and the inside diameters of the annular space.

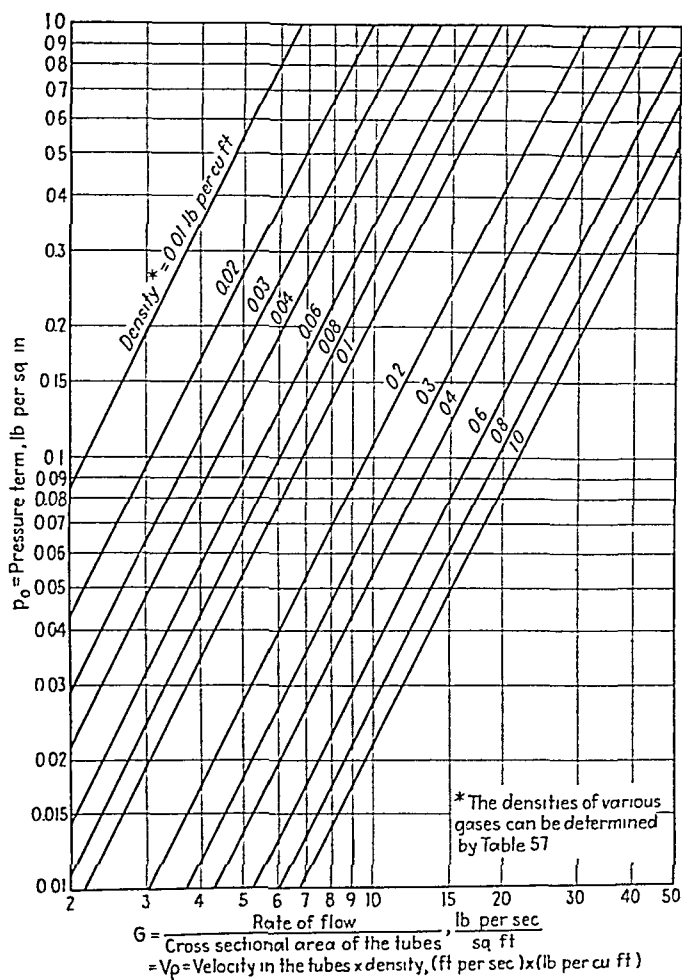


FIG 16 — Pressure term for Case 8

## CHAPTER VI

### TYPES OF INDUSTRIAL HEAT-TRANSFER EQUIPMENT

**36 Introduction**—There are many types of heat-transfer equipment. Some of the more common types used for transferring heat from one fluid to another are described in this chapter. Direct-fired or electrically heated equipment is not included.

**37 Shell-and-tube-type Equipment**—a One of the most widely used types of heat transfer equipment consists of a num-

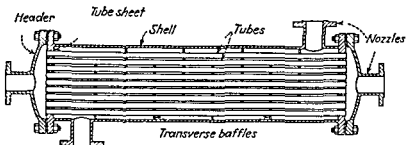


FIG. 47—Cross section of a simple shell and tube heat exchanger

ber of parallel tubes inclosed in a relatively close-fitting cylindrical shell. One fluid flows inside the tubes and is called the *tube side fluid*; the other flows outside the tubes and is called the *shell side fluid*. If neither fluid condenses or evaporates, the equipment is commonly designated as a *heat exchanger*. If one of the fluids condenses, the equipment is designated either as a *condenser* or as a *heater*, depending upon whether the primary purpose of the equipment is to condense the one fluid or to heat the other. Similarly, if one of the fluids evaporates, the equipment is designated either as an *evaporator* or as a *cooler*, depending upon whether the primary purpose is to evaporate the one fluid or to cool the other.

One of the simplest designs is shown in Fig. 47. The tube sheets are an integral part of the shell, and both fluids make a single pass through the equipment. The transverse baffles shown are not essential, but they increase the velocity and

turbulence of the shell-side fluid and thereby increase the rate of heat transfer. They are omitted if the shell-side fluid condenses or evaporates or if the volume of flow is large. The size of the nozzles used depends upon the service for which the equipment is intended. For example, the vapor-inlet nozzles on condensers intended for vacuum service are usually made quite large, are not necessarily circular in cross section, and may extend almost the entire length of the shell.

Although the design shown in Fig. 47 is simple, it has certain disadvantages. Thus, no provision is made to allow for differences in the expansion of the tubes and of the shell. Consequently, the design is not suitable if the temperature difference

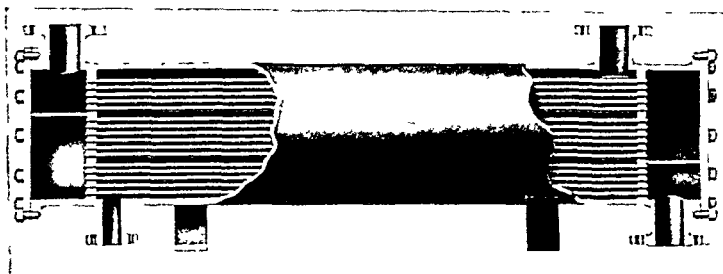


Fig. 48.—Cutaway view of a shell-and-tube heat exchanger arranged for multi-pass flow of the tube-side fluid. (Courtesy of the Griscom-Russell Co.)

between the tube-side and shell-side fluids is large. Moreover, since the outside surfaces of the tubes are not accessible for cleaning, the design cannot be used if the shell-side fluid is dirty or scale forming. Finally, the headers cannot be removed in order to clean the inside surfaces of the tubes without breaking the piping connections.

Another relatively simple design is shown in Fig. 48. This is similar to the one shown in Fig. 47 in that (1) no provision is made to allow for differences in the expansion of the tubes and of the shell and (2) the tube sheets are an integral part of the shell; so the outside surfaces of the tubes are not accessible for cleaning. This design offers a distinct improvement over the preceding one, however, in that the inside surfaces of the tubes can be readily cleaned without disturbing any of the piping connections. Note that the headers are provided with partitions

which cause the tube-side fluid to make three passes through the exchanger

Many variations of these two simple designs are possible (1) by using various kinds of tubes, (2) by using various kinds of baffles in the shell, (3) by providing partitions in the covers to produce multipass flow, and (4) by including some provision to allow for differences in the expansion of the tubes and of the shell

*b Tubes*—Straight, bare tubes are most commonly used, but for certain applications other kinds are advantageous. Straight tubes set into the tube sheets with a slight initial bow or curvature are useful when the shell-side fluid is scale forming. With such tubes changes in temperature cause variations in flexure



FIG. 43—Cross section of a shell and tube heat exchanger with U tubes

which tend to crack up and break off scale accumulations. Bowed tubes also provide for expansion and contraction but they should not be used in short lengths because of the increased likelihood of tube failure at the tube sheets.

U, or "hairpin" tubes illustrated in Fig. 49 are used to take care of expansion and contraction. These require only one tube sheet, but single defective tubes are sometimes accessible for replacement only by removing several other tubes. Since the tubes are difficult to clean internally they are suitable for use only when the tube-side fluid is clean and nonfouling.

Finned tubes are particularly adapted for use when the shell side fluid has a much lower film coefficient than the tube-side fluid, *e.g.*, when a gas is to be heated or cooled by a liquid. Either transverse or longitudinal fins, as shown in Fig. 50 can be used depending upon whether the flow is normal to or parallel to the tubes.

Retarder, or "swirling," strips, consisting of a twisted metal strip inserted inside the tubes, are rarely used. Better results can be obtained with the same pressure drop by using either

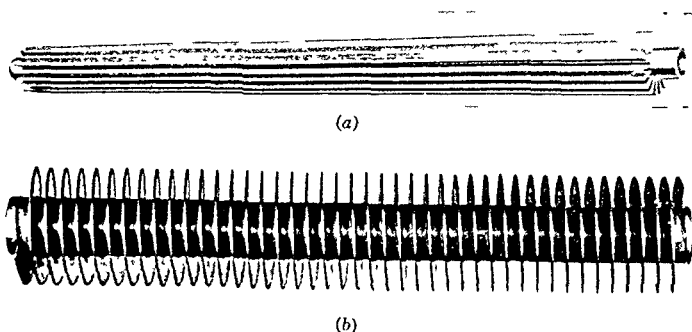


FIG. 50.—Finned tubes. (a) Longitudinal fins. (b) Transverse fins. (Courtesy of the Griscom-Russell Co.)

longer or smaller diameter tubes or by increasing the number of passes.

c. *Baffles*.—There are two general types of baffles: transverse and longitudinal. Both are used to increase the rate of heat transfer by increasing the velocity and turbulence of the shell-side fluid. They add considerably to the cost of the equipment,

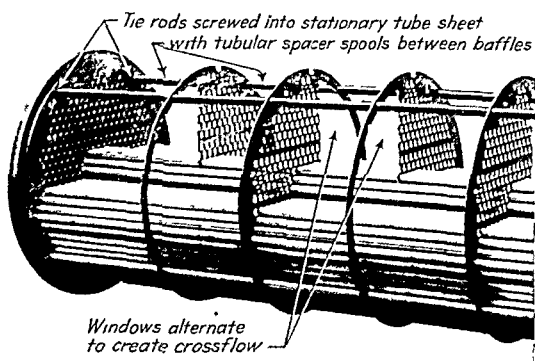


FIG. 51.—Partially assembled tube bundle showing method of holding transverse baffles in place. (Courtesy of the Ross Heater and Mfg. Co.)

however, and to the difficulty of cleaning the outside surface of the tubes. They also increase the pressure drop through the shell. In order to be fully effective, they must be installed so

that by-passing is eliminated as far as possible. This is accomplished by reducing to a minimum the clearance between the baffles and the shell and between the baffles and the tubes. Sufficient clearance must be allowed, however, to permit the tube bundle to be withdrawn readily, even after some warping has taken place. The baffles are held in place independent of the tubes by tie rods and spacers, as shown in Fig 51. It is important that the baffles be supported rigidly in order to prevent vibration which would result in chafing and quick failure of the tubes.

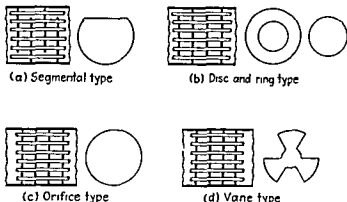


FIG. 52—Types of transverse baffles (tube holes not shown)

Several kinds of transverse baffles are used, probably the commonest being the *segmental* or "cross flow," type shown in Fig 52a. This type of baffle causes the shell-side fluid to flow back and forth across the tube bundle and is suitable for use with fouling liquids or for large volumes of flow.

*Disk-and-ring* baffles shown in Fig 52b, force the fluid to flow successively from the outside of the tube bundle toward the center and back again to the outside. These baffles are slightly more expensive than the segmental type and have no particular advantage over them.

*Orifice* baffles, shown in Fig 52c, force the fluid to flow at high velocities through the annular space between the baffles and the tubes. They can be used for small or moderate volumes of flow.

*Vane-type* baffles, shown in Fig 52d, are slightly twisted like the blades of a fan. By turning the baffles so that the blades of



one are in line with the openings of the next, a moderate deflection of the fluid is obtained. These baffles provide a large area for flow and consequently cause only a small pressure drop. They are particularly suitable for use with gases.

*Helical* baffles are made of a number of disks which are cut along one radius and welded together to form a continuous helix. They are more expensive than any of the other types and have no outstanding advantages over them.

*Longitudinal* baffles consist of flat plates which extend all the way across the shell. They may be used to obtain counter-flow of the tube-side and shell-side fluids, as shown in Fig. 53. They are also frequently used in condensers to distribute the flow of vapor over the entire length of the tube bundle when

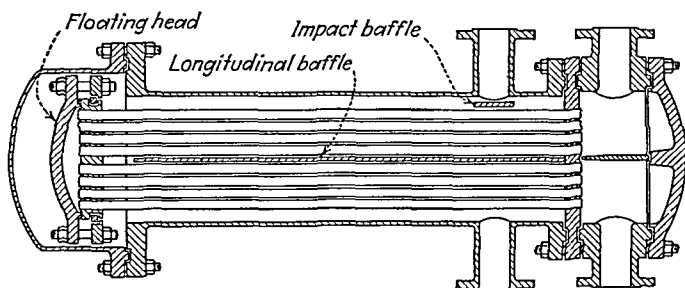


FIG. 53.—Cross section of a shell-and-tube heat exchanger equipped with a full floating head.

the vapor-inlet nozzle is located in the center of the shell. The clearance between the baffle and the shell should be as small as possible in order to prevent by-passing, but at the same time it must be large enough to permit the tube bundle to be withdrawn easily. If a split floating head is used, the longitudinal baffle may be permanently welded to the shell.

*Impact* baffles, as shown in Fig. 53, are placed opposite the shell-inlet nozzle. They are used to prevent erosion, caused by impingement of the incoming fluid on the tubes, and to prevent tube vibration. They also help to distribute the flow.

*d. Passes.*—If the tube-side fluid passes from one end of the heat exchanger to the other end just once, as in Fig. 47, the exchanger is said to be *single-pass* on the tube side. In contrast, the heat exchangers shown in Figs. 48 and 53 are said to be *three-pass* and *two-pass*, respectively. By adding suitable partitions

inside the headers, the tube-side fluid can be made to make as many passes as may be desired. Such *multipass* arrangements are used to obtain higher velocities and longer paths for the fluid to travel without increasing the length of the exchanger.

*c Expansion Provisions*—It is frequently necessary to make some provision to allow for differences in the expansion of the tubes and of the shell. It is sometimes also necessary to provide for differences in the expansion of the various tubes if either the tube-side or the shell side fluid undergoes a large change in

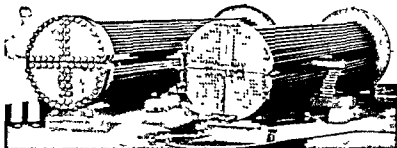


FIG. 54—Tube bundles with split floating heads (Courtesy of the Henry Vogt Machine Co.)

temperature. The following methods are used to provide for these differences in expansion.

The *full floating-head* construction, shown in Fig. 53, is relatively expensive but is particularly suited for use when the shell-side fluid is under high pressure. The tube bundle can be removed readily for cleaning the outside surface of the tubes or for making repairs, and since straight tubes are used the inside surface of the tubes can also be easily cleaned. The channel type of cover shown on the stationary head permits easy access to the inside of the tubes for cleaning or inspection without requiring the piping connections to be broken.

The *split floating-head* construction shown in Fig. 54 is used if either of the fluids passing through the exchanger undergoes a large change in temperature. The tube-side fluid is made to pass successively through each of the floating heads, and the differences in the temperatures of the tubes connected to any one floating head are thereby reduced.

The *packed floating-head* construction, shown in Fig. 55, also permits free expansion of the tubes and the inside surface

of the tubes are readily accessible for cleaning by removing the cover plates on the stationary and floating heads. The entire tube bundle can be withdrawn to allow access to the outside surfaces of the tubes. The exchanger shown is two-pass on the tube side; so one row of tubes is omitted in the center of the tube

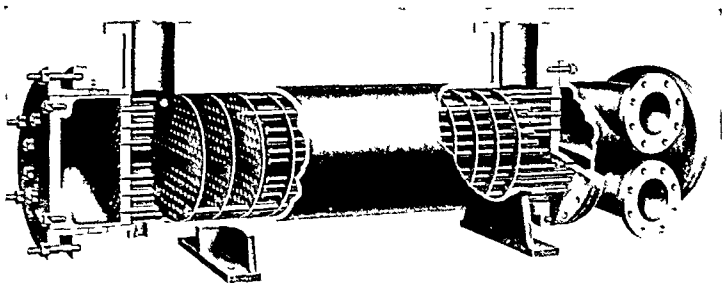


FIG. 55.—Cutaway view of a shell-and-tube heat exchanger equipped with orifice-type baffles and a packed floating head. (Courtesy of the Griscom-Russell Co.)

bundle because of the partition required in the stationary head. Note the tie rods and spacer spools used to hold the orifice-type baffles in place.

A modification of the packed floating-head construction is shown in Fig. 56. A smaller packing gland is required, and single-pass flow through the tubes is possible. This design is rarely used, however, because it is as expensive as either the full floating head or packed floating head without having any outstanding advantages over them. Note that the expansion of the tubes is transmitted to the external piping and that the latter must be arranged to take care of this movement.

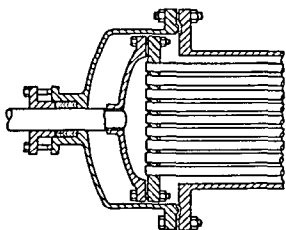


FIG. 56.—Cross section showing packing gland on pipe connected to floating head.

Differences in the expansion of the tubes and of the shell are frequently provided for by some sort of flexible section built into the shell. Three such designs are shown in Fig. 57. These designs are suitable for moderate differences in expansion.

The *U* tubes shown in Fig. 49 offer a relatively inexpensive means of providing for expansion. Since the tubes are free to

expand independently of each other, they are particularly suited for use where the various tubes may be subjected to widely differing temperatures. They can be used for any pressure, but because of the difficulty of cleaning them internally, they should not be used when the tube-side fluid is dirty or scale forming. *Bowed tubes* can also be used to take care of expansion, but

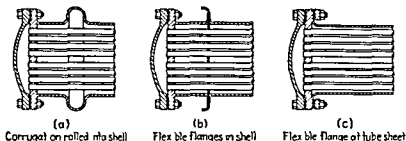


FIG. 57—Types of flexible sections built into the shell

because of the repeated flexing they are likely to fail or develop leaks at the tube sheet unless they are relatively long.

Individually packed tubes, each tube provided with a stuffing box and *ferrule* as shown in Fig. 58, are frequently used in steam power plant condensers. This arrangement allows each tube to expand independently but involves a large number of joints to be kept tight.

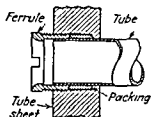


FIG. 58—Individually packed tube

Another arrangement that permits independent tube expansion consists of two concentric tubes. One end of the outer tube is plugged and the two tubes are expanded into separate tube sheets. The tube-side fluid flows through the smaller tube to the plugged end of the larger and back through the annular space between them. Heat transfer between the shell-side fluid outside the larger tube and the tube-side fluid inside takes place while the latter is in the annular space.

**38 Double-pipe Heat Exchangers**—Double-pipe heat exchangers are sometimes used in place of shell and tube heat exchangers for low rates of flow. They usually consist of two concentric pipes, one of the fluids flowing through the inner pipe and the other flowing in the opposite direction through the

annular space between the pipes. True counterflow is obtained, but this advantage is somewhat offset by the relatively large outside surface requiring insulation. On the basis of cost per square foot of heat-transfer surface, these exchangers are relatively expensive and consequently are usually used only for special applications.

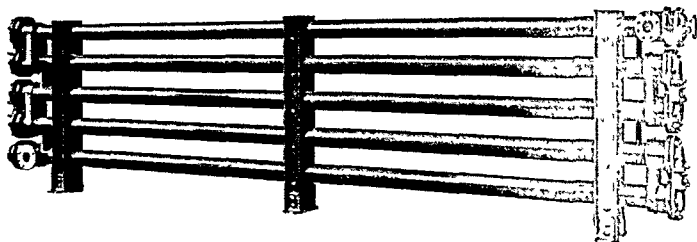


FIG. 59.—Double-pipe heat exchanger. (Courtesy of the Struthers-Wells Co.)

A typical double-pipe heat exchanger is shown in Fig. 59. Expansion is provided for by the packing gland on the inner pipe.

A special design of double-pipe heat exchanger, intended for use when a gas is to be heated or cooled by a liquid, is shown in Fig. 60. The longitudinal fins on the inner pipe compensate

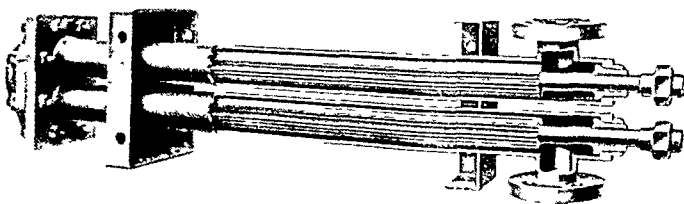


FIG. 60.—Double-pipe heat exchanger equipped with longitudinal fins. (Courtesy of the Griscom-Russell Co.)

for the low film coefficient of the gas and reduce the length of pipe required.

A rather unusual design is shown in Fig. 61. Instead of the tubes being concentric, they are expanded throughout their entire length into fins, heat flowing from one fluid to the other through the fins. The inlet and outlet connections and the return bends

type has 2-in. diameter tubes about 5 ft. long, although tubes from 1 to 4 in. in diameter and from 3 to 6 ft. long are common.

c. *Basket-type Evaporator*.—The basket-type evaporator, shown in Fig. 64, operates in the same manner as the standard vertical short-tube evaporator except that the downtake is annular. The deflector helps reduce entrainment losses and can be easily installed. One of the advantages of the basket-type

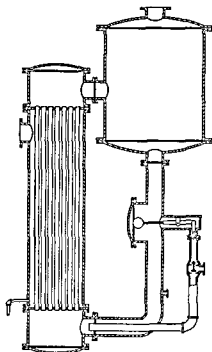


FIG. 65 —Vertical long tube evaporator

evaporator is the fact that the entire heating element can be removed for cleaning and repairing. About the same diameter and length tubes are used as in the standard short-tube evaporator.

d. *Vertical Long-tube Evaporator* —Vertical long-tube evaporators are similar to vertical short-tube evaporators, except that the tubes are usually from 10 to 20 ft. long. These evaporators can be divided into two types: the controlled-level type, shown in Fig. 65, and the film type. In the controlled-level type the liquor level is maintained at from one-third to one-half the height

of the tubes by a float-controlled feed valve in the downtake. The vapor discharges either into a vapor head directly above and integral with the heating element or into a separate vapor head as shown. Baffles are used to help separate the entrained liquor from the vapor.

In the film-type evaporator the liquor level is not controlled, and the downtake pipe is used only in starting. After the

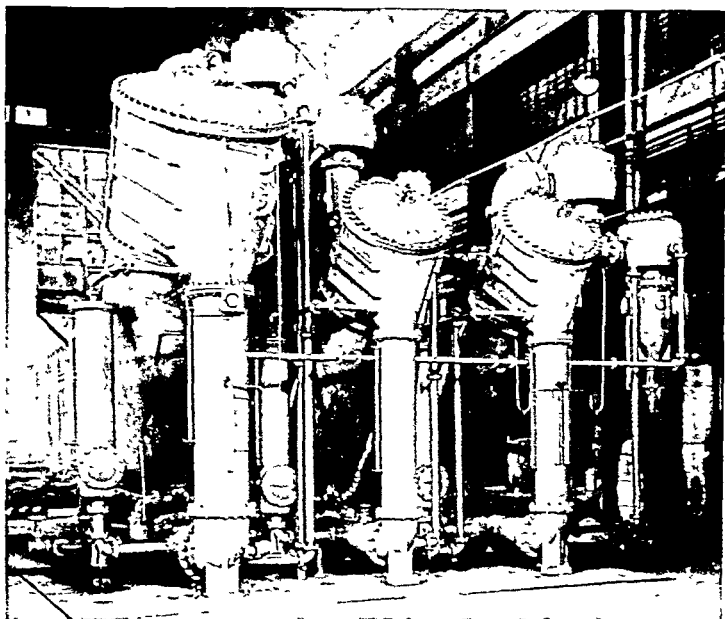


FIG. 66.—Triple-effect inclined-tube evaporator. (Courtesy of the Struthers-Wells Co.)

evaporator is in operation, the liquor makes only a single pass through the tubes, the vapor and entrained liquid discharging usually into a centrifugal-type separator. The liquid carried over into the separator is withdrawn through a float-controlled valve which is required to prevent by-passing of the vapor.

*e. Inclined-tube Evaporator.*—Inclined-tube evaporators are similar to vertical long-tube evaporators except that the tubes are inclined, usually at an angle of 45 deg. This type of evaporator requires less headroom than the vertical type, and tube replacements are more easily made. Since these evaporators can

be made self-supporting, they require no structural framework and are relatively easy to install. A large triple-effect evaporator of this type is shown in Fig. 66. Note the downtake pipes and liquid level controllers.

*f Horizontal tube Evaporators*—Horizontal-tube evaporators differ from vertical- and inclined-tube evaporators in that the

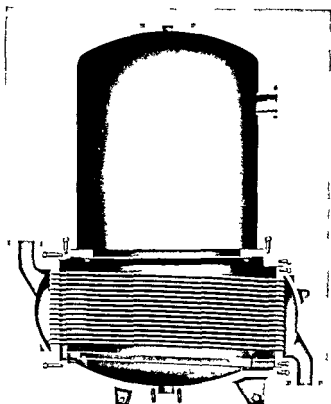


FIG. 67 —Sectional view of a vertical body horizontal tube evaporator (Courtesy of the Griscom-Russell Co.)

liquor is outside the tubes and the steam inside. They are made with either rectangular or cylindrical bodies. The rectangular type of body is often built of flat cast-iron plates and is known as the *Wellner-Jelinek type*. This construction permits easy access for repairs but involves a large number of joints to be kept tight. The cylindrical type of body is built with the body either vertical or horizontal. Figure 67 shows a vertical-body



evaporator equipped with bowed tubes which, when subjected to changes in temperature, tend to crack off and shed scale automatically. The horizontal-body evaporator shown in Fig. 68 is also equipped with bowed tubes and is designed for producing boiler-feed make-up water. The entire tube bundle can be withdrawn, and the inside surfaces of the tubes can be cleaned by unscrewing the plugs in the headers opposite the ends of each tube.

*g. Forced-circulation Evaporator.*—The forced circulation evaporator is similar to the long-tube vertical evaporator, except

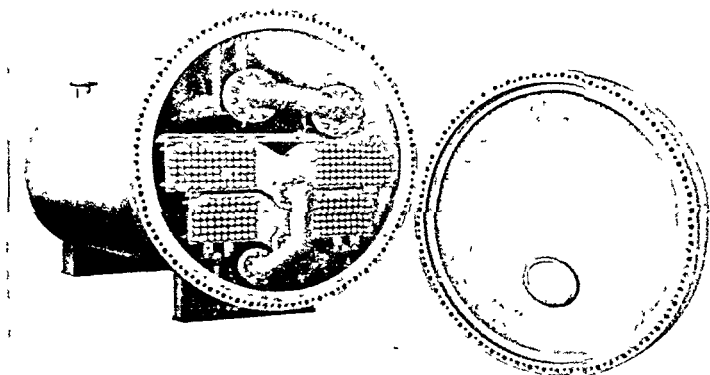


FIG. 68.—Horizontal-body, horizontal-tube evaporator with cover removed.  
(Courtesy of the Griscom-Russell Co.)

that a centrifugal pump is placed in the circuit between the downtake and the tube bundle, as shown in Fig. 69. This permits higher velocities, and consequently higher rates of heat transfer, to be obtained. Since less heating surface is required, this type of evaporator is particularly adapted to applications that require that the heating surface be made of some high-priced metal. The liquor, which is inside the tubes, emerges from the heating element at high velocities and is directed downward by a baffle into the vapor head. The feed and thick-liquor take-off connections can be made at any point in the circuit, although admitting the feed on the inlet side of the pump is advantageous in that the feed and thick liquor are thoroughly mixed on passing through the pump.

*h Coiled-tube Evaporator*—Another type of evaporator intended for producing boiler feed make up water is shown in Fig 70. The heating surface consists of helical coiled tubes which, like bowed tubes, crack up and break off scale accumulations automatically when subjected to variations in temperature.

**41. Open-type Sections**—Open-type sections are characterized by the fact that the tube bundle is not inclosed in any kind

of shell. A typical open-type section is shown in Fig 71. These sections are made with either bare or finned tubes and can be arranged for multipass flow of the tube-side fluid by providing partitions in the headers.

Bare-tube sections are frequently used as condensers or as gas or liquid coolers. For either of these applications, the sections can be submerged in open tanks or pits through which the cooling water flows. Baffles may be provided in the tanks to obtain counter-flow of the cooling water. If a limited water supply makes atmospheric cooling necessary, the sections can be suspended directly beneath cooling towers, and the cooling water allowed to drip over them.

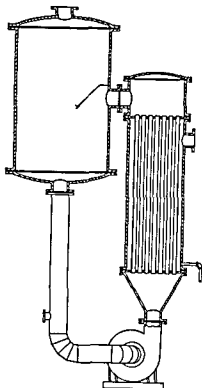


FIG 69—Forced-circulation evaporator

sections are subjected to large variations in temperature or if the cooling water is scale forming.

Bare-tube sections are often used for recovering heat from hot waste water in plants, such as dye houses, woolen mills, bleacheries, and laundries. The sections are placed in pits through which the hot waste water flows while the clean water to be heated flows through the tubes. This arrangement is particularly advantageous if the waste water contains lint or other

solid matter. Such impurities would quickly clog the passages of a shell-and-tube heat exchanger, but they can easily be washed off open sections with a hose.

Bare-tube sections are sometimes also used for heating chemicals or other solutions. The sections are submerged in the solution, and the steam for heating flows through the tubes.

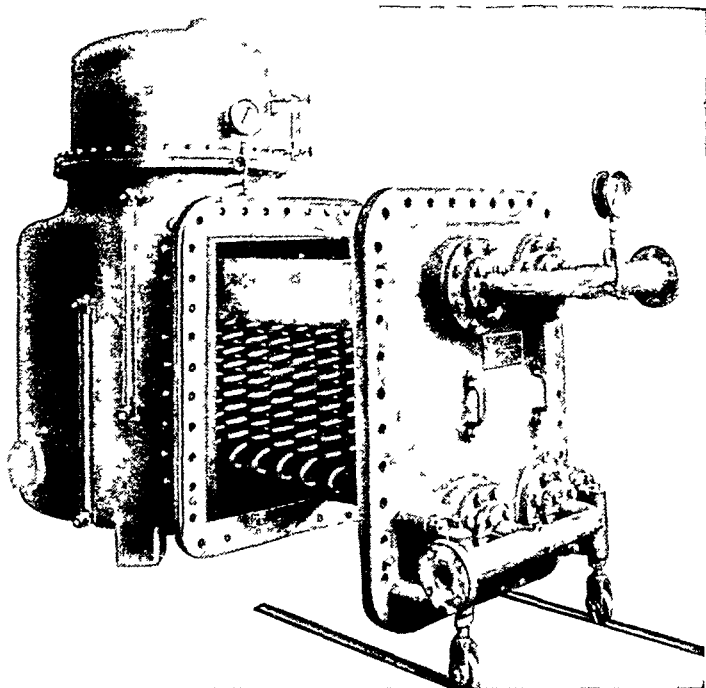


FIG 70—Coiled-tube evaporator (Courtesy of the Griscom-Russell Co)

Finned-tube sections are used for heating or cooling large volumes of gas at atmospheric pressure. The gas is blown across the finned surface, while either steam or some cooling medium flows through the tubes. For example, Fig. 72 shows a finned-tube section intended for use as a generator air cooler. The air flows from the cooler to the generator and back in a closed circuit and is cooled by cooling water flowing through the tubes of the cooler.

**42 Pipe Coils**—Pipe coils are an inexpensive type of heat-transfer equipment and are made in innumerable forms. Two of the simpler forms are shown in Fig. 73. Pipe coils are used

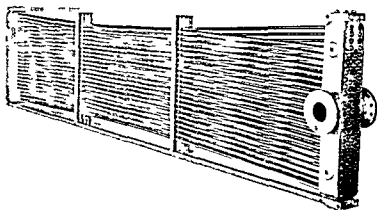


FIG. 71—Open type section with bowed tubes (Courtesy of the Grascom-Russell Co.)

as gas or liquid coolers, the coils being submerged in cooling water; they are used as solution heaters, the heating steam

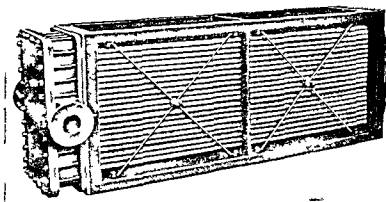


FIG. 72—Open type section with finned tubes (Courtesy of the Grascom-Russell Co.)

flowing through the coils, and they are used as refrigerant evaporators, the refrigerant evaporating as it flows through the coils.

43. Hot-water Heaters.—The necessity of producing large quantities of hot water arises so frequently that several kinds

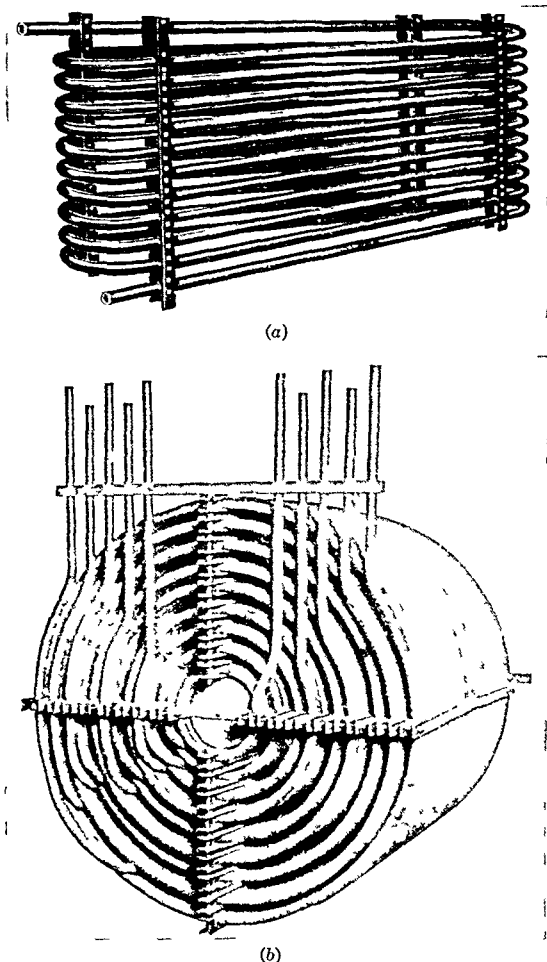
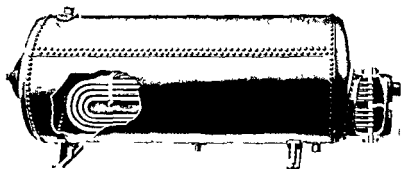


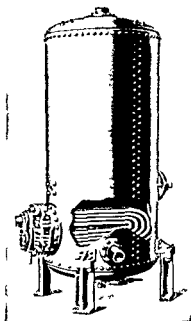
FIG. 73 —Typical pipe coils (a) Oblong coil. (b) Closely nested helical coils with all terminals at one end. (Courtesy of the Whillock Coil Pipe Co.)

of equipment designed especially for this purpose are manufactured. These heaters use steam as the heating medium and are of two types: storage heaters and instantaneous heaters.

Storage heaters are intended for use where large quantities of hot water are required at irregular intervals. They consist



(a)



(b)

FIG 74 —Hot-water storage heaters (a) Horizontal type (b) Vertical type  
(Courtesy of the Patterson-Kelley Co)

of a hot-water storage tank equipped with a bundle of U tubes, as shown in Fig 74. These heaters are sometimes made with

two separate tube bundles, one for high-pressure steam and the other for exhaust steam.

Instantaneous heaters differ from storage heaters in that the water to be heated flows through the tubes while the heating steam is in the shell. A typical instantaneous heater of the shell-and-tube type is shown in Fig. 75. Another design using coiled tubes is shown in Fig. 76. These heaters are particularly adapted for use where the demand for hot water is fairly steady.

**44. Miscellaneous Types of Equipment.** *a. Jacketed Vessels.* Jacketed vessels are frequently used for boiling liquids and are usually heated by steam condensing in the jacket. The heat-

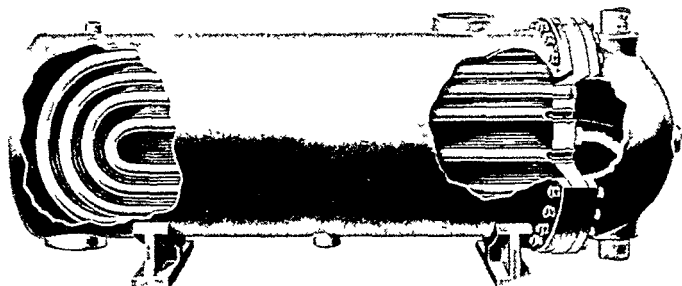


FIG. 75.—Instantaneous hot-water heater equipped with U tubes. (Courtesy of the Patterson-Kelley Co.)

transfer surface in contact with the liquid being boiled is relatively small but can be easily cleaned. For this reason jacketed vessels are used in place of submerged tube bundles for applications, such as cooking, that require frequent and thorough cleaning of all surfaces.

*b. Submerged Steam Jets.*—If the addition of some water is not objectionable, liquids can be heated by a submerged jet of steam. The rate of steam flow should be regulated so that all the steam is condensed and none escapes from the surface. By directing the jet properly, the steam can also be made to help circulate the liquid.

*c. Sprays.*—Gases can be very efficiently heated or cooled by a spray of liquid if saturating the gas with the liquid vapor is not objectionable. This system is sometimes used, for example,

in air conditioning installations. The air to be cooled flows through sprays of cold water and, because of the large amount of water surface, is cooled very nearly to the temperature of the water.

*d Regenerative type Equipment*—Regenerative-type heat exchangers differ from all the previously mentioned types in that the heat to be transferred is stored within the exchanger for a period of time. These exchangers usually consist of a chamber filled with a checkerwork of firebrick, cast iron or other material capable of absorbing considerable heat, and they operate in a cyclic manner. The hot fluid flows through the exchanger first and gives up heat to the checkerwork. The cold fluid then flows through the exchanger and absorbs this heat. Exchangers of this type are regularly used for preheating air for blast furnaces and open hearth furnaces.

#### 45 Tube Sizes and Materials

The choice of tube size usually depends upon the service for which the equipment is to be used and upon economic considerations. Smaller diameter tubes and closer spacing allow more surface to be installed in a given size tube bundle. Also, for the same velocity of flow, film coefficients in small tubes are higher than in large. Larger tubes

FIG. 76—Instantaneous hot-water heater equipped with coiled tubes. (Courtesy of the Gracom-Russell Co.)

are easier to clean, however, and wider spacing allows cleaning lanes for mechanical cleaning of the outside surface. Regarding tube-wall thickness, thinner tubes are cheaper and are easier to expand into the tube sheets, but they wear out faster. Regarding tube lengths, fewer tubes are required if longer tubes are used, and consequently cleaning and repairing are easier. The



most frequently used tube sizes together with certain useful geometric data are listed in Table V of the Appendix.

Different materials vary in their ability to resist each of the various factors affecting the life of tubes. Since many combinations of these factors occur, tubes made of a number of different materials are necessary. Some of the more common tube materials are low-carbon steel, stainless steel, copper, brass, Admiralty, Muntz, copper-nickel alloys, aluminum, and lead.

**46. Tube-installation Methods.**—The commonest method of installing tubes consists of expanding the ends of the tubes into the tube-sheet holes by means of a rotating, roller expanding tool,

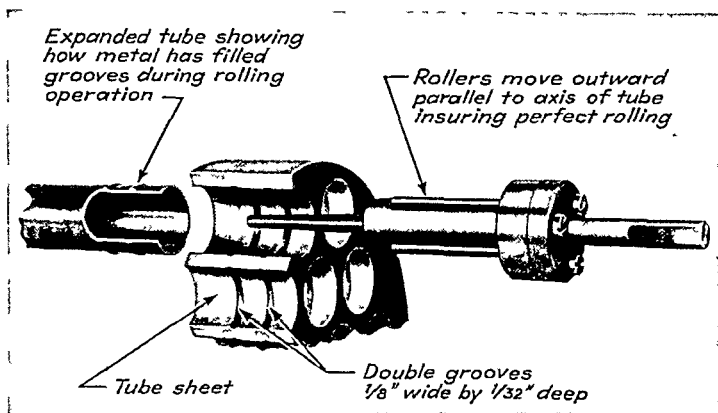
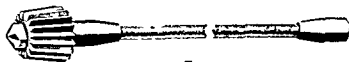


FIG. 77.—Method of expanding tubes into tube sheet. (Courtesy of the Ross Heater and Mfg. Co.)

as shown in Fig. 77. The tube-sheet holes are drilled undersize and are then reamed to a diameter slightly larger than the outside tube diameter. One or more grooves are frequently cut inside the holes in order to reduce the possibility of leakage and to increase the structural strength of the joints. Care must be taken not to overroll the tubes, since this would reduce the thickness of the tube walls at the point where corrosion and erosion are most severe. The inlet ends of the tubes are sometimes flared in order to reduce the likelihood of a contraction of the stream taking place. Expanding tubes by drifting (driving a conical pin into the ends of the tubes) is unsatisfactory and is seldom done.

Tubes are also frequently installed with ferrules, as shown in Fig 58. This method of installation permits easy tube replacements but has the disadvantage of involving a large number of joints to be kept tight.

Tubes are sometimes sweated or soldered into the tube sheets. This method is particularly adapted to the installation of small-diameter tubes which would be difficult to expand. It can also be used if the tubes are spaced so closely that expanding the



a



b



c

FIG 78 —Scale cutters for small tubes (a) Cone cutter with flexible connection for cleaning U tubes (b) Expanding-blade head and drill head (c) Cone cutter universal joint and steam or air-driven motor (Courtesy of the Firetool Mfg Co)

tubes might damage the tube sheet. The method cannot be used if the equipment is to be used at high temperatures.

Tubes can also be welded into the tube sheets, but this method of installation makes tube replacements very difficult.

**47 Cleaning of Tubes** —Cleaning the inside surfaces of tubes is facilitated by providing the headers with covers, such as those shown in Figs 48, 55, and 72, that can be removed without breaking any of the pipe connections. Removable plugs opposite the ends of each tube, as shown in Fig 71, also allow easy access

to the inside of the tubes. Cleaning the outside surfaces is facilitated by removable tube bundles and wide spacing of the tubes and baffles.

Hard, scaly deposits inside the tubes can be removed by means of cutters, such as those shown in Fig. 78. These cutters are driven by air-, steam-, or water-operated motors and are usually provided with a universal joint or some other type of flexible connection to permit their being used in bowed or U tubes. Soft or gummy deposits inside tubes can be removed by motor-driven revolving brushes or by rubber plugs forced through the tubes by water pressure.

Hard, scaly deposits on the outside surfaces of tubes can be removed by sandblasting, if the tube bundle is removable, or by motor-driven vibrators or knockers which are run through the inside of the tubes. Soft, gummy deposits can be removed with hand brushes. If the tube surfaces are not readily accessible, certain types of deposits can sometimes be dissolved by means of chemical cleaning fluids.

## CHAPTER VII

### INSULATION

**48 Introduction**—Any material having a low thermal conductivity can be used as insulation. A good insulating material must possess certain other properties, however. If it is to be used at high temperatures, it must be able to withstand these temperatures without deteriorating. If it is to be used at low temperatures, it must be either protected from or unaffected by moisture. For some applications the material may need to have sufficient structural strength to be self-supporting, it may need to be tough enough to resist rough usage and vibration, and it may need to be fireproof or resistant to chemical action. For other applications the material may need to be light in weight, odorless, and verminproof. In addition to these various factors, the choice of the kind of insulating material as well as the thickness to be used depends upon the cost. Other factors that must be considered in this connection are the ease of application, the salvage value of the insulation, and the cost of the heat or of the refrigeration that the insulation will save.

**49 Basic Insulating Materials**—Practically all industrial insulating materials are made from the following basic materials: asbestos, magnesium carbonate, diatomaceous silica, vermiculite, rock wool, glass wool, cork, cattle hair, and wool.

*Asbestos*, a mineral possessing a crystalline, fibrous structure is obtained from asbestos mines in the form of fine, silky crystals. These crystals are easily separated by successive crushings into individual fibers. In order to be of commercial value, the asbestos must possess length and fineness of fiber combined with infusibility, toughness, and flexibility.

*Magnesium carbonate* is produced from dolomite limestone through a series of chemical processes including calcination of the dolomite, slaking in water, the addition of carbon dioxide to eliminate calcium carbonate, and precipitation of the magnesium carbonate. Magnesium carbonate owes its low con-

ductivity to the great number of microscopic dead air cells that it contains.

*Diatomaceous silica*, also called *diatomaceous* or *infusorial earth*, consists of the siliceous skeletons of microscopic plants called *diatoms*. These diatoms live in water; and when they die, their skeletons fall to the bottom of the sea or lake and build up deposits of diatomaceous earth. Such deposits have been formed in the past and have been hardened into solid rock, the individual boxlike skeletons remaining unaltered, however. These earths are generally white or gray in color and are soft and friable.

*Vermiculite* is a micaceous mineral which possesses the peculiar characteristic of expanding in one direction only on being heated. The expansion takes place at right angles to the plane of cleavage and may amount to as much as sixteen times the original volume.

*Rock wool* and *glass wool* are made by melting silica minerals in a furnace and pouring the molten mass in fine streams into a high-velocity jet of steam. The molten mass is blown into silky fibers which rapidly cool and solidify. The resulting woollike formation is chemically inert and incombustible and has a low thermal conductivity because of the many void spaces between the fibers.

*Cork* is the outer layer of the bark of an evergreen oak tree known as the *cork oak*. This bark becomes unusually thick and is stripped off every 8 or 10 years. Cork is cellular in structure and owes its low thermal conductivity to the air entrapped within the cells.

**50. Pipe Covering for Hot Lines.**—*a.* Insulation for covering hot pipe lines is available in the form of half sections, whole sections, segmental blocks, blankets, and tape.

*Half sections* are the most commonly used form of insulation for pipe sizes up to about 10 in. They are held in place by metal bands or by loops of soft iron wire spaced from 9 to 12 in. apart. The ends of the wire are twisted, bent over, and hammered into the insulation so as to leave no projections. For extra-thick insulation, the half sections are applied in two layers, the circumferential and the longitudinal joints being staggered.

*Whole sections* can be used only when the insulation is sufficiently flexible to allow its being slipped over the pipe when split longitudinally along one side. This method of application is

advantageous in that it increases the effectiveness of the insulation by decreasing the number of joints

*Flat blocks* or sets of *curved segmental blocks* are used for insulating pipes larger than 10 in. These blocks are held in place by soft iron wire loops and a thin coat of hard finish asbestos cement is applied over them to present a smooth even surface

When any of the foregoing types of rigid insulation are used the insulation is stopped off a sufficient distance from all flanges so that when the ends are beveled back at an angle of 45 deg the flange bolts may be easily removed

*Blankets* consisting of some insulating material held between two sheets of metal lath are applied by wrapping them around the pipe and lacing the edges of the metal lath together with wire

*Insulating tape* is sometimes used when the application of sectional insulation is impractical. It is wrapped spirally around the pipe and held in place by wire clamps

*b* 85% *Magnesia* is the most commonly used kind of insulation for temperatures up to 600°F. It has a low thermal conductivity, is light in weight, is easily cut and fitted, is not affected by steam or water leakage, and is sufficiently strong to withstand all ordinary usage. It consists of a mixture of approximately 85 per cent magnesium carbonate and 15 per cent asbestos fiber. The asbestos fiber is added to provide the necessary bonding and reinforcing qualities to allow the material to be molded into half sections and blocks, as shown in Fig. 79. The half sections are 3 ft. long and vary in thickness from  $\frac{3}{8}$  in. to 1 $\frac{1}{2}$  in. depending upon the pipe size. The flat or curved segmental blocks are usually either 3 by 18 in. or 6 by 36 in. in size and are from  $\frac{1}{2}$  to 4 in. thick.

The method of finishing depends upon the location. On piping located indoors a canvas jacket may be sewed on over rosin sized paper, or in the case of the half sections the flaps on the light canvas jacket, which is regularly furnished as an integral part of the sections, may be pasted down and the joints covered with brass lacquered bands. On piping located outdoors the insulation may be covered with a jacket of roofing paper, the joints all being lapped over and sealed with asphalt cement. On horizontal pipes the seams of the jacket are placed at the side of the pipe with the lap turned down in order to shed water. The jacket is secured in place by copper wire loops spaced about 4 in.

apart. Where the insulation is in danger of mechanical injury, a metal jacket may be used.

c. *Calcined diatomaceous silica and asbestos fiber*, mixed with a bonding material, are used in the manufacture of an insulation that is suitable for temperatures up to 1900°F. The diatomace-

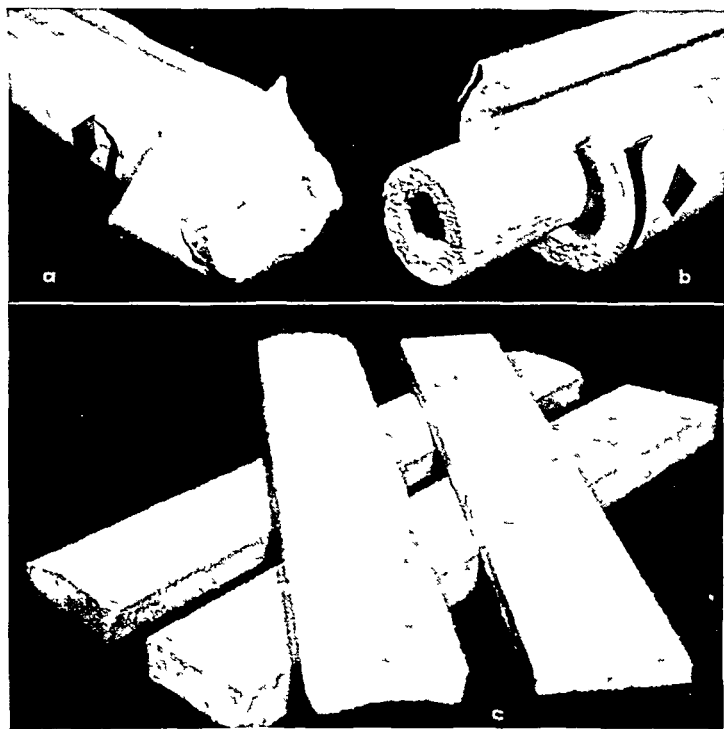


FIG 79—85% Magnesia insulation (a) Single-layer pipe insulation (b) Double-layer pipe insulation (c) Blocks and lagging. (Courtesy of the Keesbey & Mattison Co)

ous silica is calcined, or burned, at a temperature of approximately 2200°F. before being mixed with the asbestos fiber, in order to limit the shrinkage of the finished product in service. This type of insulation does not have so low a thermal conductivity as 85% Magnesia but can be used at much higher temperatures. For this reason it is frequently used as an inner layer in combination with 85% Magnesia, as shown in Fig. 5 in Chap. I.

A sufficient thickness is used so that the temperature of the latter does not exceed 600°F the maximum allowable temperature for 85% Magnesia. The greater insulating efficiency of the 85% Magnesia is thus combined with the higher heat resistance of the high temperature insulation to produce a covering that has a maximum efficiency for a given thickness of insulation. This combination also reduces the cost of the insulation since 85% Magnesia is less expensive than the high temperature insulation.

This type of insulation is molded into half sections 3 ft long from 1 to 3 in thick for pipe sizes from  $\frac{1}{4}$  to 10 in. For larger pipe sizes it is ordinarily furnished in segmental blocks.

*d Laminated asbestos felt* insulation is especially suited to applications involving excessive vibration or rough usage. It consists of approximately 40 layers per inch of thickness of a felt made of asbestos fiber and particles of finely ground sponge. It has an unusually low thermal conductivity because of the great amount of dead air entrapped between the layers of felt and it possesses considerable toughness because of the felted nature of the material. It is more expensive than molded insulations such as 85% Magnesia because it must be fabricated. It may be used for temperatures up to 700°F.

Sections for pipe covering are manufactured by winding the layers of felt one over the other and cementing them together at intervals. The resulting cylinder is then split lengthwise to permit application. For pipe sizes 3 in. and larger it is split on one side only since the material is sufficiently flexible to permit the sections to be slipped on to the pipe as shown in Fig. 80. For pipe sizes smaller than 3 in. the cylinder is cut into half sections. The sections are 3 ft in length and are available in thicknesses from 1 to 3 in. For service indoors the sections are supplied with canvas jackets and are held in place with brass lacquered bands. For service outdoors they are supplied with integral waterproof asbestos jackets.

*e Felted asbestos* insulation is also suitable for applications involving vibration expansion and contraction. It can be used for example for covering gas engine exhaust lines. It is made from asbestos fibers felted and molded into half sections and can be used for temperatures up to 1000°F. It is made in 3 ft sections from 1 to 3 in thick to fit standard pipe sizes up to 12 in.



*f. Felted rock wool* secured between two layers of metal lath by tie wires, as shown in Fig. 81a, is used for temperatures up to 1000°F. Because of its flexibility, this type of insulation is shipped flat and is applied by wrapping it around the pipe and lacing the edges of the lath together with wire. For indoor work the insulation is coated with asbestos cement and finished with a

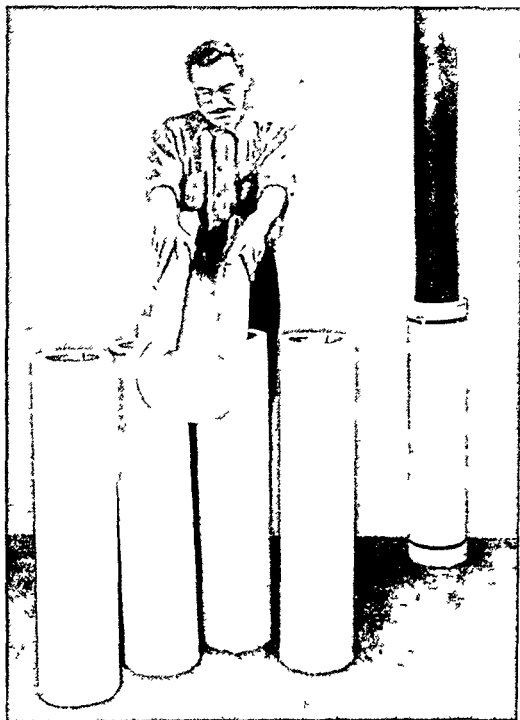
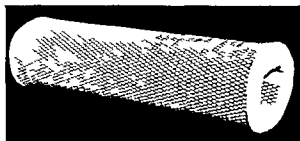


FIG. 80—Laminated asbestos-felt pipe insulation. (Courtesy of the Johns-Manville Co)

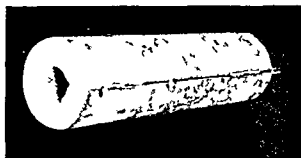
canvas jacket. For outdoor work it is finished either with a coat of weatherproof cement or with a galvanized-iron outer casing, as shown in Fig. 81b. The latter is particularly suited to installations where a fireproof and weatherproof jacket is required or where the insulation must be frequently removed for repairs, changes, or inspection. This type of insulation is made in sections 2 ft. long, in thicknesses from 1 to 4 in., for pipe sizes

2 in in diameter and larger Special shapes for flanges and bends can be obtained

*g* Glass wool supported by straight-line wire netting shown in Fig 82 is used for covering pipes 8 in in diameter and larger at temperatures up to  $1000^{\circ}\text{F}$  Since the wire netting is on the outside face only this type of insulation is particularly adapted for covering spirally riveted or spirally welded pipe Rods



(a)



(b)

FIG 81—Felted rock wool pipe insulation (a) Secured between metal lath (b) Equipped with galvanized iron outer casing (Courtesy of the Johns-Manville Co.)

bound to the edges of the wire netting form a selvage which facilitates lacing the covering together The sections are made 2 ft long in thicknesses from 1 to 4 in

*h* Quilted asbestos blankets shown in Fig 83 are made of asbestos cloth filled with loose asbestos fiber and are used on pipes where frequent inspection requiring the removal of the insulation is necessary These blankets are easier to apply than molded insulation and are not subject to breakage They are provided with metal hooks and are laced on to the pipe with



FIG. 82.—Glass-wool pipe insulation. (*Courtesy of the Owens-Corning Fiberglas Corp.*)



FIG. 83.—Quilted asbestos blankets. (*Courtesy of the Johns-Manville Co.*)

wire. They may be used for temperatures up to 900°F and are manufactured in sections 3 ft long in thicknesses from  $1\frac{1}{2}$  to 3 in.

1 *Insulating tape*, consisting of loosely twisted strands of asbestos cloth, can sometimes be used where lack of space or numerous bends make the application of rigid, sectional insula-



FIG. 84.—Insulating tape. (Courtesy of the Johns-Manville Co.)

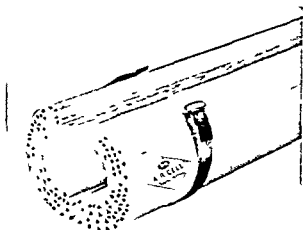


FIG. 85.—Corrugated asbestos pipe insulation. (Courtesy of the Keasbey & Mattison Co.)

tion impractical. The tape can be used for temperatures up to 400°F and will withstand considerable vibration and rough usage. It is wrapped spirally around the pipe and is held in place by wire clamps as shown in Fig. 84. It is made in 50-ft. rolls, in width up to  $2\frac{3}{4}$  in., and in thicknesses up to  $\frac{3}{8}$  in.

2 *Corrugated asbestos insulation* shown in Fig. 85 may be used for temperatures up to 300°F. It is an inexpensive type of

insulation and is regularly used for covering low-pressure steam-heating lines. It consists of alternate layers of plain and corrugated asbestos paper, cemented together at the points of contact, and is made in half sections which are hinged together by either a canvas or a heavy asbestos-paper jacket. The sections are held in place on the pipe by pasting down the flap on the canvas jacket or by inserting metal staples into the paper jacket. The circumferential joints are covered with metal bands.

This type of insulation is made in sections 3 ft. long, in thicknesses from two to eight plies, to fit all standard pipe sizes. The standard ply is  $\frac{1}{4}$  in. thick, although finer corrugations averaging  $\frac{1}{6}$  in. per ply are also made.

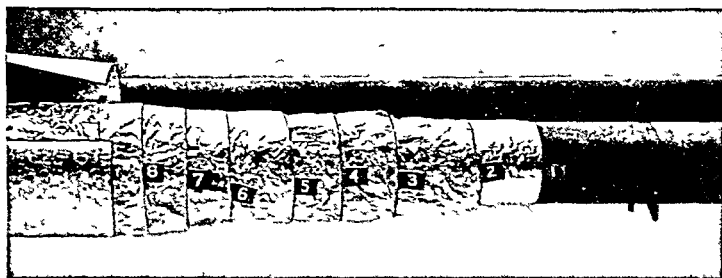


FIG. 86.—Aluminum-foil pipe insulation. (Courtesy of the Alfol Insulation Co)

k. *Wool-felt* insulation, designed particularly for use on either hot- or cold-service water lines, may be used for temperatures up to 225°F. It consists of an inner liner of waterproofed asbestos felt covered with a continuous wrapping of creped-wool felt, a wrapping of asbestos paper over the wool felt, and a canvas jacket pasted over the asbestos paper. The asbestos paper is added to make the insulation match other insulation in appearance, wool felt itself being gray in color and darker than either asbestos or magnesia. The insulation is split longitudinally to permit application, the canvas jacket and outer layer of felt acting as a hinge. The sections are held in place by pasting down the canvas flap, the joints between the sections being covered with metal bands. This type of insulation is manufactured in sections 3 ft. long, in thicknesses of  $\frac{1}{2}$ ,  $\frac{3}{4}$ , and 1 in. for all pipe sizes from  $\frac{1}{2}$  in. to 5 in.

l. A comparatively recent development is the use of *crumpled aluminum foil* for insulating pipes. This type of insulation has

a very low thermal conductivity, is unusually light in weight, and has a very low heat storage capacity. The aluminum foil used is approximately 0.0003 in. thick and is furnished in smooth rolls. It is crumpled by hand before being wrapped around the pipe and is applied loosely enough that the air spaces between the sheets are about  $\frac{3}{8}$  in. thick. Since the foil has no structural strength itself, it must be protected from crushing or injury by a covering of rigid material such as galvanized iron or pressed asbestos, as shown in Fig. 86. Note the wire frames used to support the galvanized iron cover. The aluminum foil will

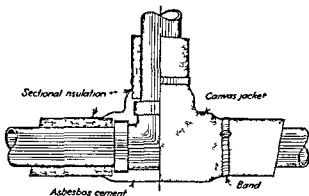


FIG. 87 — Asbestos cement insulation for small fittings and valves. (Courtesy of the Johns-Manville Co.)

withstand considerable vibration and can be used for temperatures up to 1000°F.

**51 Valves and Fittings on Hot Lines** — Valves and fittings on hot lines in the smaller pipe sizes ( $3\frac{1}{2}$  in. or smaller) are usually insulated with hard finish asbestos cement as shown in Fig. 87. Successive layers of cement, each approximately  $\frac{1}{4}$  in. thick, are applied until the total thickness is equal to that of the insulation on the adjacent piping. Each layer is allowed to dry thoroughly before the next layer is applied, the final layer being troweled to a smooth, hard finish.

On the larger pipe sizes (4 in. or larger) the bodies of flanged fittings and the entire surface of screwed fittings are covered with block insulation as shown in Fig. 88. The blocks are usually made of the same material as that used on the adjacent piping and are held in place by loops of soft iron wire. They are applied to a thickness  $\frac{1}{2}$  in. less than that of the insulation

on the piping, and hard-finish asbestos cement is applied over them to make the total thickness of insulation equal to that on the piping.

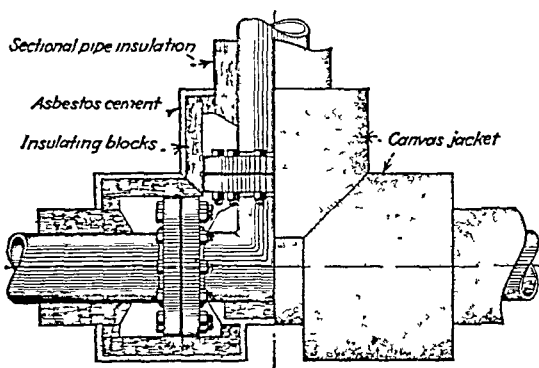


FIG 88 —Block insulation for large fittings and valves (Courtesy of the Johns-Manville Co)

Flanges may be insulated in either of two ways: If the insulation is to be permanent, blocks long enough to extend about 2 in. over the adjacent pipe insulation are wired around the

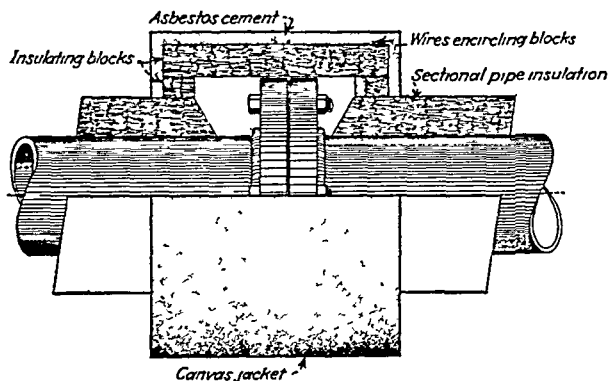


FIG. 89.—Permanent type of insulation for flanges (Courtesy of the Johns-Manville Co)

flange and are finished with a coating of asbestos cement, as shown in Fig. 89. If the insulation is to be removable and replaceable, either sectional or block insulation can be used to

construct a split cover. If sectional insulation is used the half sections must be large enough to encircle the flange completely and long enough to extend about 2 in. over the pipe insulation as shown in Fig. 90. The sections are held in place with wire,

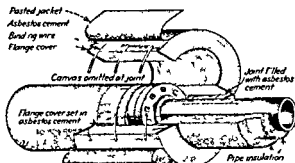


FIG. 90—Removal and replaceable sectional insulation for flanges (Courtesy of the Johns Manville Co.)

and the joint between the flange insulation and the pipe insulation is sealed with asbestos cement. A smooth even surface is obtained by applying a thin coat of cement over the entire cover. If the cover is made of blocks the blocks are wired to a frame

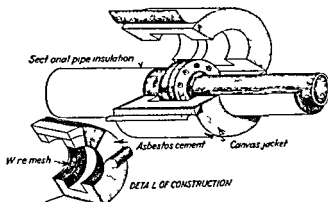


FIG. 91—Removable and replaceable block insulation for flanges (Courtesy of the Johns Manville Co.)

made of  $\frac{1}{2}$  in. square mesh galvanized wire cloth as shown in Fig. 91. The mesh is split at the ends and bent down to inclose asbestos cement used to fill the annular space between the blocks and the pipe insulation. The cover is wired in place



on the flange, and the entire surface is covered with a coat of asbestos cement.

The insulation on valves and fittings located indoors is finished with a covering of canvas, cut and stretched to cover the entire

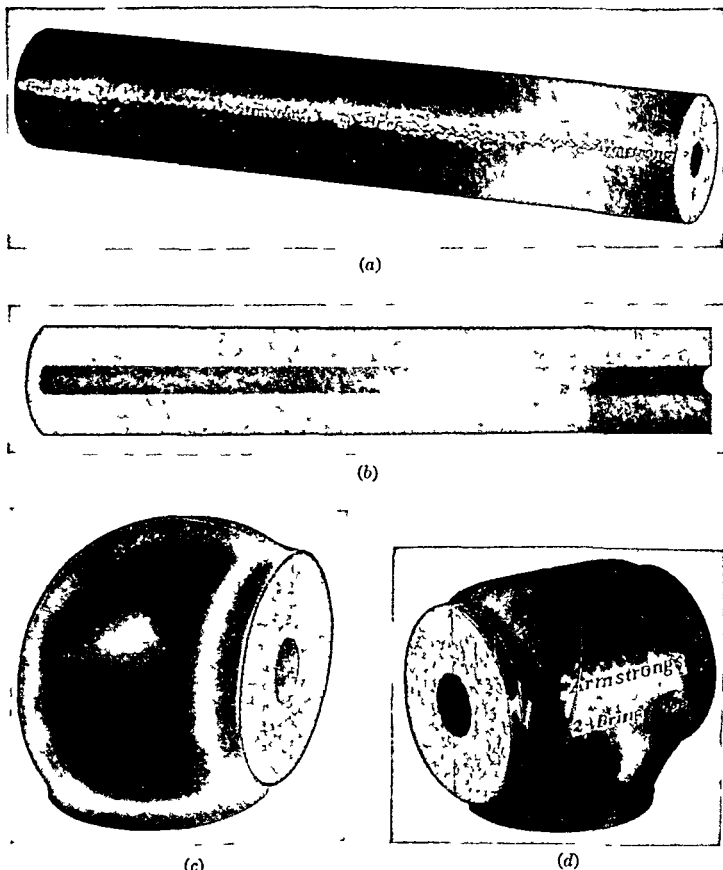


FIG 92 —Typical sections of cork insulation. (a) Full section of pipe covering. (b) Half section of pipe covering. (c) Screwed ell cover. (d) Screwed tee cover. (Courtesy of the Armstrong Cork Co.)

surface and held in place by paste. If the insulation is exposed to the weather, it is finished by applying a  $\frac{1}{4}$ -in. coat of cement consisting of two parts asbestos fiber and one part Portland

cement This is covered in turn with a  $\frac{1}{4}$  in coat of weather proof asphalt cement

**52 Pipe Covering for Cold Lines**—*a* In order to be permanently effective, the insulation on all pipes colder than the surrounding atmosphere must be finished with a surface that is practically airtight Otherwise, the insulation will eventually become completely water soaked by the moisture condensed out of the air filtering into the insulation This will not only destroy the effectiveness of the insulation but on freezing may burst and damage it

*b* Cork insulation is manufactured from granules of pure cork compressed into molds under high pressure and baked The baking process liquefies the natural resin in the cork, which

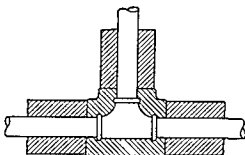


FIG 93—Cross section of cork covering for screwed tee

cements all the particles into a homogeneous mass The inside surface of the insulation is machined to form a snug fit on the pipe and the outside surface is finished with a heavy mastic coating Sections are molded to fit practically all standard screwed and flanged fittings as well as to fit straight pipe Some typical sections are shown in Fig 92

The insulation is held in place by loops of copper clad steel wire all the joints being coated with waterproof cement as the insulation is applied Screwed fittings are usually covered first The sections for covering these are machined to form a snug fit as shown in Fig 93 Straight pipes are covered next the sections being cut long enough to be wedged slightly against the covers on the screwed fittings or against the bolt heads of the flanged fittings Flanged fittings are covered last The sections for covering these do not fit snugly on to the fitting but rest on the adjacent pipe insulation as shown in Fig 94 The

annular space between the fitting and the sectional covering is filled with a mixture of melted paraffin and granulated cork. This mixture is poured through a hole bored through the sectional covering at the highest point and is allowed to solidify. On indoor lines the insulation is usually finished with a coat of asphalt paint; on outdoor lines it is covered with a layer of roofing paper.

Cork insulation is made in three thicknesses known, respectively, as *Special Thick Brine*, *Brine Thickness*, and *Ice Water Thickness*. *Special Thick Brine* varies in thickness from 2.63 to 4.00 in., depending upon the pipe size, and is intended for lines

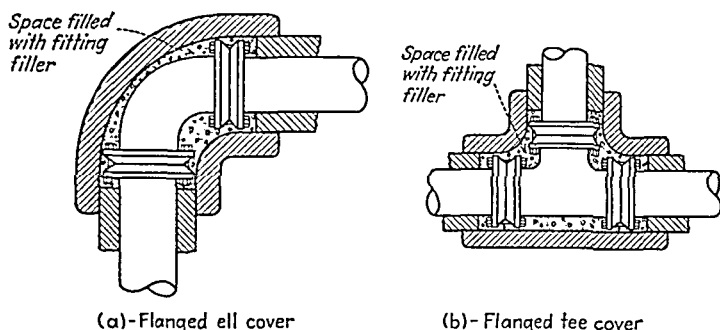


FIG. 94.—Cross section of cork covering for flanged fittings. (Courtesy of the Armstrong Cork Co.)

carrying fluids at temperatures between 0 and  $-35^{\circ}\text{F}$ . For temperatures below  $-35^{\circ}\text{F}$ ., the thickness of the insulation is increased by applying segmental blocks (lagging) over the sectional covering. *Brine Thickness* varies from 1.70 to 3.00 in. and is intended for temperatures between 0 and  $35^{\circ}\text{F}$ . *Ice Water Thickness* varies from 1.20 to 1.93 in. and is intended for temperatures above  $35^{\circ}\text{F}$ . All three thicknesses are made in sections 3 ft. long to fit standard pipe sizes up to 8 in. The insulation is made in the form of segmental blocks for larger pipe sizes.

c. Loose rock wool impregnated with a waterproof binder is molded into half sections and segmental blocks for covering straight pipe. The half sections are made with an integral waterproof jacket which acts as a hinge, as shown in Fig. 95. The sections are held in place by cementing down the flap with waterproof cement and twisting several loops of wire around each

section The ends of the sections are sealed against the infiltration of air by a coating of waterproof cement, applied as the sections are slipped into place, and by wrapping the joints with waterproof tape

The sections are made in 3 ft lengths in *Heavy Brine*, *Brine*, and *Ice Water* thicknesses The *Heavy Brine* thickness varies from 2 82 to 4 00 in, depending upon the pipe size, and is intended for covering pipes carrying fluids at temperatures between 0 and  $-30^{\circ}\text{F}$  It is made in half sections for pipe sizes up to 6 in and in segmental blocks for larger sizes For tem



FIG. 95—Method of applying molded rock wool pipe covering (Courtesy of the Johns Manville Co.)

peratures below  $-30^{\circ}\text{F}$  the thickness of the insulation is increased by applying segmental blocks over the *Heavy Brine* covering *Brine* thickness varies from 1 97 to 3 19 in and is intended for temperatures between 0 and  $30^{\circ}\text{F}$  It is made in half sections for pipe sizes up to 8 in and in segments for larger sizes *Ice Water* thickness varies from 1 4 to 2 0 in and is intended for temperatures above  $30^{\circ}\text{F}$  It is made in half sections for pipe sizes up to 10 in and in segments for larger sizes

Valves and fittings are insulated with loosely felted rock wool impregnated with asphalt This material is made in rolls 18 in wide with a nominal thickness of approximately  $1\frac{1}{2}$  in The

felt is applied by cutting a piece from the roll two or three times the circumference of the fitting, folding it double or triple according to the thickness desired, and placing it around the fitting. It is secured in place with jute twine and sealed against the infiltration of moisture by a wrapping of waterproof tape, as shown in Fig. 96. On the larger sizes a second and third layer of the felt are applied, each layer being sealed with tape.



Fig. 96 —Method of insulating valves and fittings with rock wool (Courtesy of the Johns-Manville Co)

The insulation is finished with a coat of waterproof cement troweled on over the tape.

*d. Hair felt* is made from cattle hair, chemically cleaned and felted. It is made in rolls (see Fig. 97) 50 ft. long, in widths from 3 to 9 ft., and in thicknesses from  $\frac{1}{4}$  to 2 in. It is a generally useful material for low-temperature work.

Hair felt can be used in place of either cork or rock wool for covering refrigerant lines by building up a suitable number of layers of the felt. The required length of felt for each layer is wrapped around the pipe and held in place by twine. Water-

proof tape is wrapped spirally around the felt and waterproof cement is applied over the tape. The insulation is finished with a canvas jacket sewed on over a layer of rosin-sized sheathing paper. This type of covering requires considerable skill and care in its application and consequently is not commonly used.

Water pipes are frequently protected from freezing by a covering made of hair felt. For severe conditions a suitable number of layers of the felt (see Sec. 63) are bound on to the pipe with twine and are covered with a canvas or a weatherproof jacket. Precaution against the infiltration of air are not needed, since the pipe is warmer than the surrounding air, and therefore

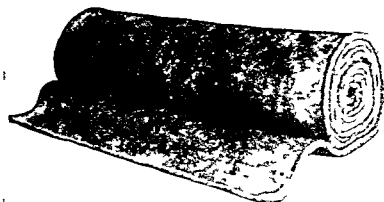


FIG. 97.—Roll of hair felt. (Courtesy of the Johns-Manville Co.)

condensation of moisture in the insulation cannot take place. If the pipes are exposed only to moderately low temperatures, or if the time that the surrounding air is below the freezing point is of short duration, a covering designed especially for these conditions can be used. This covering shown in Fig. 98, consists of two thicknesses of hair felt protected on both the inside and outside faces by several layers of waterproofed wool felt. The sections are split longitudinally to permit application, the outer layer of wool felt acting as a hinge. They are held in place by pasting down the flap on the canvas jacket and covering the joints with metal bands. The sections are made in 3-ft. lengths to fit all pipe sizes from  $1\frac{1}{2}$  in. up but in only one thickness approximately equal to  $1\frac{1}{4}$  in.

Hair felt is also used for covering valves and fittings on lines insulated with either hair or wool felt. The felt is held in place with twine and is usually sealed against moisture either with a wrapping of waterproof tape and a coat of asphaltic paint or with a  $\frac{1}{4}$ -in. layer of asbestos cement.

*e. Wool felt* is used in the construction of two types of insulation intended for covering cold-water pipes. The first type, described in Sec. 50k, is made of creped-wool felt. The second

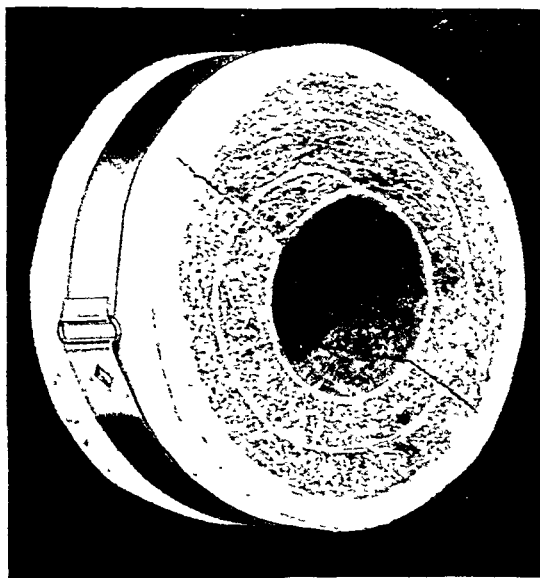


FIG. 98.—Hair-felt insulation for protecting water pipes from freezing. (Courtesy of the Johns-Manville Co.)

type, shown in Fig. 99, is intended for more severe operating conditions and is made of flat-wool felt protected from moisture by layers of waterproofed felt. It is made in single-layer construction for thicknesses up to  $\frac{3}{4}$  in. and in double-layer construction for thicknesses up to 2 in. The sections are 3 ft. long and are made to fit standard pipe sizes up to 6 in.

These two types of insulation serve not only to keep the water in the pipes cold but also to prevent condensation on the outside of the pipe. The prevention of condensation is of particular importance in industries where dripping from pipes might

interfere with the manufacturing process or might damage finished goods. The thickness of insulation required (see Sec. 64) depends upon the temperature and relative humidity of the air surrounding the pipe and upon the temperature of the water in the pipe, and it must be such that the temperature of the outside surface of the insulation is above the dew point of the air.

53. **Block Insulation for High Temperatures.**—*a.* Large surfaces, both flat and curved, can be covered with insulation in

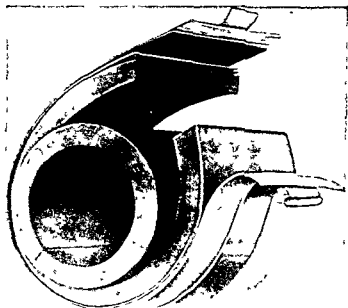


FIG 99—Wool-felt insulation for covering cold-water pipes (Courtesy of the Johns-Manville Co.)

the form of blocks. These blocks are made of the same materials that are used for making pipe covering. Blocks made of 85% *Magnesia* are the most commonly used and are suitable for temperatures up to 600°F. For temperatures up to 1900°F., blocks made of *calcined diatomaceous silica* and *asbestos fiber* are used. Both these kinds of blocks are made either flat or curved to any desired radius, in widths of 3, 6, or 12 by 18 or 36 in. long, in thicknesses from 1 to 4 in.

Blocks made of *laminated asbestos felt* can be used for applications involving excessive vibration or rough usage. They can



be used for temperatures up to 700°F and are made in widths from 6 to 36 by 36 in. long, in thicknesses from  $\frac{1}{2}$  to 4 in. They are made flat but are sufficiently flexible to permit their being applied to curved surfaces. Blocks made of *felted asbestos* can also be used where the insulation is subjected to vibration or to compression and strain. These blocks are 6 in wide by 36 in. long and from  $\frac{1}{2}$  to 4 in thick. They can be used for temperatures from 500 to 900°F, depending upon how well they are supported

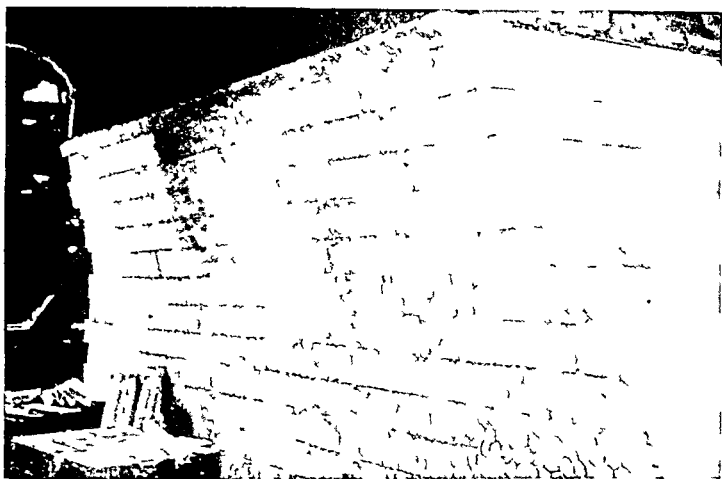


FIG 100.—85 % Magnesia blocks wired to flue and ready for final coat of asbestos cement (Courtesy of the Johns-Manville Co)

b. The usual method of applying block insulation is illustrated in Fig. 100. It consists in wiring the blocks on to the surface, stretching either hexagonal-mesh wire netting or expanded metal lath over the blocks, and applying one or more layers of asbestos cement over this. The blocks may be held temporarily in place during erection by an *adhesive cement* (see Sec. 58).

One method of wiring the blocks on to flat metal surfaces is shown in Fig. 101. Drilled angle clips are spot-welded to the metal surface, and the tie wires are threaded through them. If the blocks are to be applied to flat wooden surfaces, strips of wood the same thickness as the blocks are nailed to the wall, and the tie wires are stapled to these

The method of anchoring the tie wires used to hold block insulation in place on tanks drums and towers is illustrated in Figs 102 and 103. The blocks are held in place on the sides

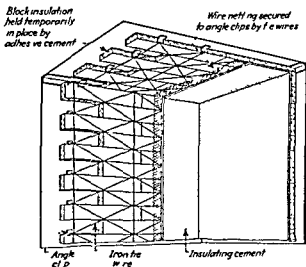


FIG. 101—Method of wiring block insulation to flat metal surfaces (Courtesy of the Johns-Manville Co.)

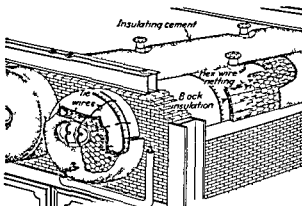


FIG. 102—Method of insulating drums and drum heads (Courtesy of the Johns-Manville Co.)

of the vessels by loops of wire which pass completely around the insulation. If the blocks are not to encircle the vessels completely, angle irons are welded along the sides of the vessels where the insulation is to terminate and the tie wires are fastened

on to these. The tie wires for holding the blocks in place over the ends of the vessels are anchored on to pencil-rod rings which are fastened around the rims of the vessels and around any pipe connections or manhole covers, as shown. These rings are held in place by clips spot-welded on to the vessels. The insulation

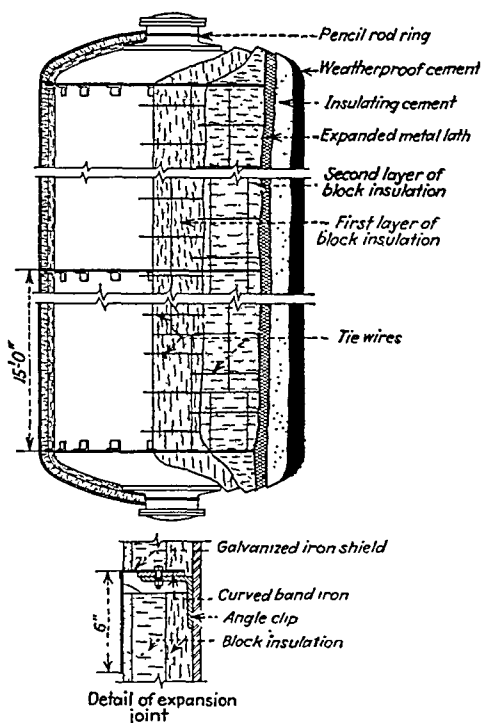


Fig. 103.—Method of insulating vertical tanks and towers. (Courtesy of the Johns-Manville Co.)

on vertical tanks and towers is supported at intervals of not more than 15 ft. by horizontal band-iron rings bolted on to angle clips, as shown in Fig. 103. These rings permit the inclusion of expansion joints and also serve as anchors for the tie wires.

In place of tie wires and wire netting, insulating blocks can be supported by some sort of rigid casing. Figure 104 illustrates this method of support applied to the insulation of a steam-boiler waterwall. Insulating cement is first applied to a thickness just

sufficient to provide a flat surface, and the blocks are held in place on this surface by the casing. In other applications the shell is erected first, and the insulation is built up inside it.

In special cases where the temperatures do not exceed 500°F and where the application of tie wires or other means of support is impractical, insulating blocks may be held in place by adhesive cement alone.

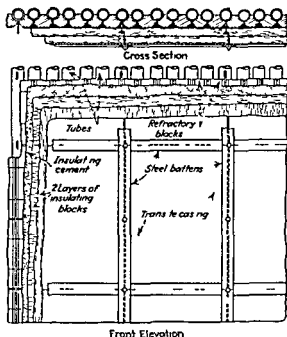


FIG. 104 Insulating blocks supported by a rigid casing (Courtesy of the Johns-Manville Co.)

**54 Block Insulation for Low Temperatures**—a Large surface subjected to temperatures below atmospheric can be covered with blocks made either of cork or of rock wool impregnated with a waterproof binder. These blocks are manufactured by the same processes used in making the corresponding types of sectional pipe covering (Secs 52b and 52c). Flat surfaces are covered with relatively wide blocks and cylindrical surfaces are covered with narrow blocks commonly known as *lags*. These lags have beveled edges to permit them to fit snugly. Flat disks are made for covering the ends of tanks.

*Cork* blocks for covering flat surfaces are made in widths from 12 to 36 in., in 36-in. lengths, and in thicknesses from 1 to 6 in. For ordinary service they are made without any special coating, but for exceptionally severe conditions they are made with an asphalt mastic coating ironed on under pressure. Cork lags are made 36 in. long, from  $1\frac{1}{2}$  to 6 in. thick, and are coated on both sides with a mastic finish. Disks are made from  $1\frac{1}{2}$  to 6 in. thick and any diameter desired. *Rock-wool* blocks are made



FIG. 105.—Showing the application of three layers of cork blocks to the walls and floor of an ice cream hardening room. (Courtesy of the Armstrong Cork Co.)

18 in. wide by 36 in. long and from 1 to 4 in. thick. Rock-wool lags are made 18 in. long, from  $1\frac{1}{2}$  to 4 in. thick, and from 2 to 5 in. wide, the width depending upon the diameter of the cylindrical surface to be covered. Disks are made from  $1\frac{1}{2}$  to 4 in. thick.

b. The blocks may be fastened to brick, stone, or concrete walls by setting them in either Portland-cement plaster or hot asphalt. Asphalt has the advantage of being waterproof but cannot be used on rough walls unless a leveling coat of Portland-cement plaster is first applied. The blocks are usually held in

place on floors and roofs by means of a mopping of hot asphalt. They are fastened to frame structures and other wooden surfaces by nailing and they are fastened to flat metal surfaces such as rectangular ducts, by means of either hot asphalt or waterproof cement. If two or more thicknesses of blocks are to be applied to any surface, each additional layer can be fastened to the preceding one by means of hot asphalt reinforced with wood skewers. As shown in Fig. 105, the successive layers of blocks are staggered, so that the joints between the blocks in any one layer are covered by the blocks in the adjacent layers.

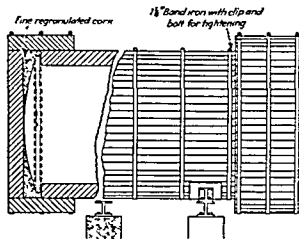


FIG. 106.—Method of applying cork lagging to cylindrical tanks. (Courtesy of the Armstrong Cork Co.)

Figure 106 illustrates the method used in applying cork lagging to cylindrical tanks. The edges of the lags are coated with waterproof cement as they are applied, a rope and spring being used to hold them in place during erection. The lags are clamped permanently in place by No. 16-gauge galvanized iron bands  $1\frac{1}{4}$  in. wide, spaced approximately 12 in. apart. The ends of the tanks are covered with disks and the spaces between the disks and the tank are filled with a mixture of granulated cork and paraffin. Rock wool lagging is applied in a similar manner except that either loose or granulated rock wool is used to fill the irregular spaces at the ends of the tank. Horizontal tanks should preferably be supported on saddles outside the insulation.

The saddles should be large enough to support the tank without compressing the insulation

Low-temperature block insulation may be finished by painting, applying hot asphalt, or applying a coat of Portland-cement plaster. The completed insulation on two brine coolers and the piping connected to them is shown in Fig. 107. The insulation is finished with a coat of aluminum paint.

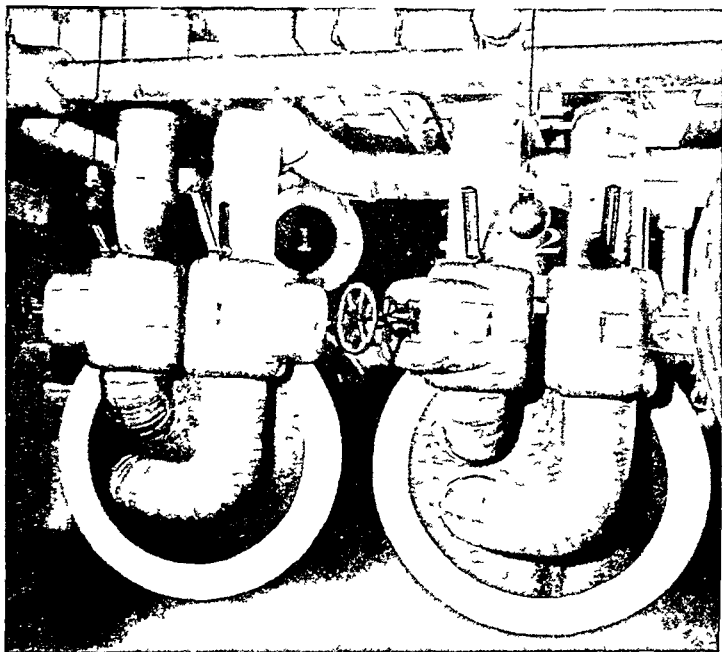


FIG 107.—Cork insulation on two brine coolers and the connected piping  
(Courtesy of the Armstrong Cork Co)

**55. Sheets.**—Sheet insulation is made in a variety of forms. The simplest is *asbestos paper*, which is made either plain or corrugated. Asbestos paper is used for insulating furnace pipes, for protecting inflammable surfaces from heat, and as a fire retarder. *Asbestos millboard* is also used as a protection against fire, heat, or acid fumes and is frequently used as a fireproof lining for floors and partitions. It is made in sheets 42 by 48 in. and in thicknesses from  $\frac{1}{32}$  to  $\frac{3}{4}$  in. Various grades are made,

some suitable for temperatures as high as 1800°F, though the most commonly used grade is not generally recommended for temperatures above 400°F

Another type of sheet insulation, similar to but less dense than asbestos millboard, is made from *felted asbestos fiber* formed into sheets 24 by 36 in in size and from  $\frac{1}{2}$  to 4 in thick. This material is particularly suitable for use where the insulation is subject to vibration or to strain due to expansion and contraction, such as in boiler-tube doors. It can be used for temperatures from 500 to 900°F, depending upon how well it is supported

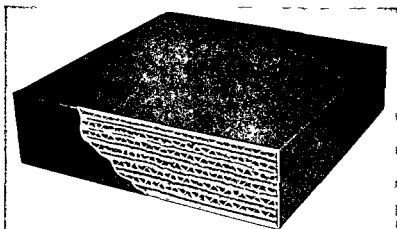


Fig 108 —Corrugated asbestos sheets protected by a refractory coating (Courtesy of the Johns Manville Co)

An inexpensive type of sheet insulation suitable for temperatures up to 300°F, is made from alternate layers of *plain and corrugated asbestos paper* cemented together at the points of contact. These sheets are used for covering low pressure boilers, dry rooms, warm air ducts, and other surfaces whose temperatures are not extreme. The sheets are made in widths from 6 to 36 in, in 36- and 72-in lengths, and in thicknesses from  $\frac{1}{2}$  to 4 in. These sheets are also made with a protective refractory coating on the outside surface as shown in Fig 108. This coating allows them to withstand temperatures up to 500°F and makes them suitable for building drying ovens, japanning ovens, and other similar heated inclosures. If the sheets are vitrified throughout, they can be used at temperatures up to



700°F. This type of sheet is made both flat and curved, as shown in Fig. 109, and is used for lining steel stacks and flues.

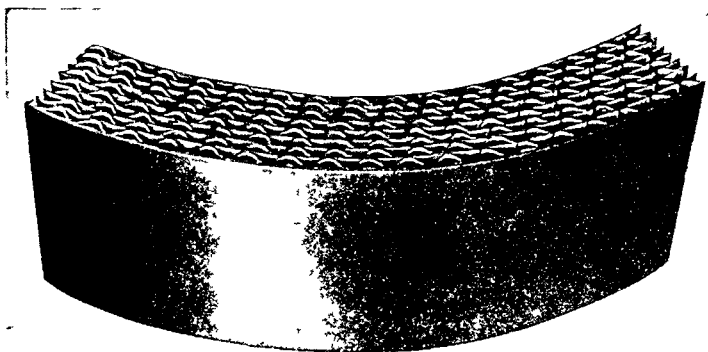


FIG. 109.—Corrugated asbestos sheets vitrified throughout (Courtesy of the Johns-Manville Co.)

A sheet insulation intended particularly for use in railroad passenger cars is made of one or more plies of *hair felt* approximately  $\frac{1}{4}$  in. thick. The individual plies are faced with kraft paper and are stitched together on approximately 5-in. centers. The assembled sheet consists of the desired number of plies inclosed in a layer of asbestos paper and a layer of muslin gauze and held together by stitching between the stitching on the individual plies. This type of insulation is made in widths from 36 to 54 in., in thicknesses from one to four plies, and in approximately 100-ft. lengths.

Sheet insulation for use in refrigerator cars is also made from *hair felt*, but waterproof paper is used in place of the kraft paper, and the insulation consists of a single ply. It is made in any width up to 108 in., in thicknesses from  $\frac{1}{2}$  to 2 in., and in practically any length desired.

Sheets of *aluminum foil* are used both as high- and as low-temperature insulation. The foil, approximately 0.0003 in. thick, is furnished in smooth rolls and is crumpled by hand before

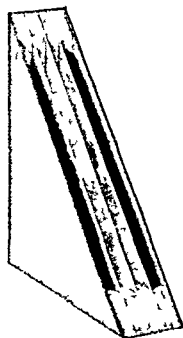


FIG. 110.—Sectional view of an aluminum-foil panel (Courtesy of the Alfol Insulation Co.)

being applied. The low thermal conductivity of this insulation is due largely to the air spaces between the sheets, the optimum thickness of air space being about  $\frac{3}{8}$  in. Since the foil has no structural strength itself, it must be protected from injury by a suitable jacket. When used as low temperature insulation, it must in addition be protected from the infiltration of air or moisture. A thin coat of lacquer is sometimes applied to the foil to protect it from corrosion. Outstanding features of this

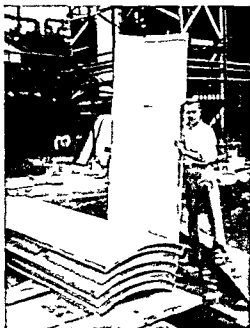


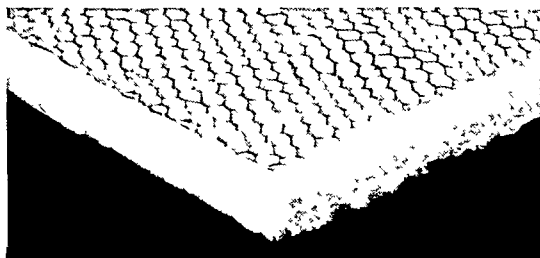
FIG. 111 — All metal aluminum foil panels. (Courtesy of the Alfol Insulation Co.)

insulation are its extremely light weight (approximately 3 oz per cu ft) and its low heat-storage capacity.

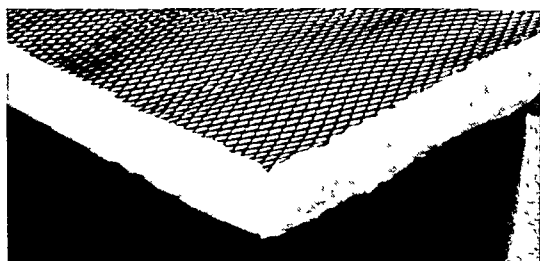
Panels made of sheets of smooth aluminum foil separated by asbestos strips illustrated in Fig. 110 are used for a number of purposes. All metal panels illustrated in Fig. 111 consist of an outer sheet metal jacket, an expanded metal mesh basket welded to this jacket, and a filling of crumpled aluminum foil sheets. These panels are suitable for covering tanks, fractionating towers, and other large flat or curved surfaces. They may be used for temperatures up to 1000°F.



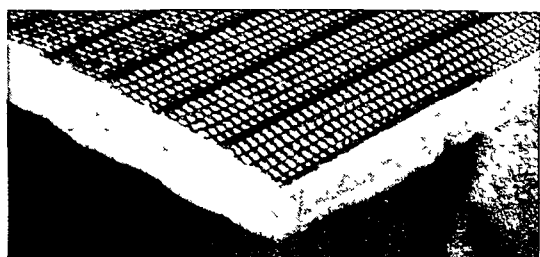
(a)



(b)



(c)



(d)

FIG. 112.—Glass-wool insulating blankets. (a) Metal facing on one side only. (b) Facing of hexagonal-mesh wire netting. (c) Expanded metal-lath facing. (d) Rib-lath facing. (Courtesy of the Owens-Corning Fiberglas Corp.)

**56 Blankets**—Large flat or curved surfaces such as those on ovens, oil refinery equipment, boilers, breechings, and hot air ducts are frequently covered with insulating blankets. The size of these blankets is more than five times the size practical for pieces of molded insulation, so fewer joints are required and faster application is possible. In addition, the edges of the blankets knit together so that there are no loose joints.

Insulating blankets may consist of felted *rock wool* or *glass wool* secured between various fabrics as shown in Fig. 112.

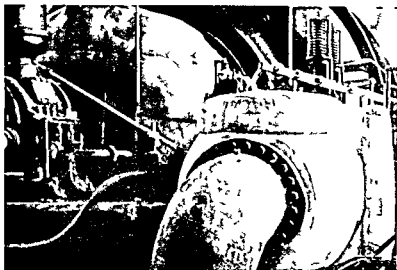


FIG. 113—Asbestos blankets used for the removable portions of turbine insulation. (Courtesy of the Johns-Manville Co.)

The most commonly used fabrics are galvanized wire netting, expanded metal lath, and rib lath. Blankets with a metal facing on one side only are used on surfaces having irregularities such as rivet heads, bolts, or laps. Galvanized wire netting is used as a facing where considerable flexibility is required for bending or fitting to irregular shapes. Expanded metal lath is somewhat more rigid than wire netting and offers an excellent base for insulating or finishing cement. Rib lath is used where maximum rigidity is required and is available with the ribs turned either in or out. Turning the ribs out provides an air space between the insulation and the hot surface, an advantage utilized in insulating steel breeching shells. The same type of facing is

not always used on both sides of the blankets, various combinations of facings being available. These blankets are used for temperatures up to 1000°F. and are made in standard sizes of 2 by 4 ft. and 2 by 8 ft., in thicknesses from 1 to 6 in.

Insulating blankets are also available with muslin, canvas, asbestos-paper, or kraft-paper facings. Muslin or canvas facings are used on blankets for insulating busses and railroad cars. Kraft-paper facings are used on blankets for insulating equipment exposed to excessive moisture, such as refrigerator cars. These types of blankets are available in any length, in widths up to 10 ft., and in thicknesses up to 4 in.

Another type of blanket used where readily removable insulation is desired consists of a quilted asbestos-cloth envelope filled with loose asbestos fiber and tufted with copper wire. These blankets are used for temperatures up to 900°F. on irregular surfaces, such as steam-turbine casings, paper-digester shells, or valve bonnets. Figure 113 illustrates the use of such blankets for the removable portions of the insulation on a steam turbine.

**57. Fillers.**—Insulating fillers are of two types: those in powder or granular form which can be poured and those in the form of loose fibers which must be packed into place by hand. Either of these types is adapted for use where structural strength and rigidity are not required of the insulation or where lack of space prohibits the use of any solid form of insulation.

Natural *diatomaceous silica* in powdered form can be poured readily and will withstand temperatures up to 1600°F. It is particularly suitable for use between refractory linings and steel, concrete, or brick shells. It is also used for covering the tops of heated equipment such as ovens, furnaces, or kilns. In such applications it is usually tamped lightly into place to prevent subsequent settling, the tamping being facilitated by moistening the powder. To prevent dusting, a lime or cement slurry is used to form a top crust. Powdered diatomaceous silica is also calcined for use at temperatures up to 2000°F.

Loose *rock wool* or *glass wool* may be used where the insulation can be packed into place. A high density of packing to prevent settling from vibration is not necessary because of the natural resilience of these materials. Rock and glass wool are also made in granulated form for use where the insulation must be poured into place. Both the loose and the granulated forms

have an advantage over powders in that they are free from dusting and do not sift through cracks. They can be used for temperatures up to 1000°F.

Other fillers that are used for high-temperature work are loose asbestos fiber, mixtures of powdered diatomaceous silica and asbestos fiber, and expanded vermiculite. For low temperature applications, ground cork, loose rock wool impregnated with asphalt, and granulated rock wool treated with a bituminous binder are used.

**58 Cements**—Insulating cements are generally used for filling cracks and finishing surfaces that are covered with molded insulation or for covering small irregular surfaces. Certain types can be used, however, for the construction of a monolithic insulation when the shape or inaccessibility of the equipment makes the use of sheets or blocks impractical. The various cements all consist of some insulating material used as a base and mixed with a suitable binder. These cements are furnished dry and must be mixed with water.

*85% Magnesia cement is the same material used in making molded 85% Magnesia insulation, except that it is in powdered form. It is a better insulator than other cements but is not ordinarily recommended either as a finishing cement or for insulating large areas. It can be used for temperatures up to 600°F.*

*Asbestos cements consist of asbestos fiber mixed with a binding material. Several grades are available. Those which produce a smooth, hard surface are used principally as a finish over sheet or block insulation. Those which are more porous are used for sealing joints and for insulating irregular surfaces, such as valves, flanges, and other pipe fittings. These cements can be used for temperatures up to 1000°F.*

*Rock or glass wool forms the base for a cement that is frequently used in place of sheets or blocks to construct a monolithic insulation, as shown in Fig. 114. If the temperature does not exceed 1200°F, this insulation can be removed, remixed with water, and reapplied. If reapplication is not essential, it can be used up to 1500°F. A reasonably smooth finish can be obtained by troweling.*

*Expanded vermiculite is the base for another cement that can be used to construct monolithic insulations. This cement is*

not quite so good an insulator as the rock- or glass-wool cements, but it can be used at higher temperatures. It can be taken off, remixed, and reapplied if the temperature does not exceed 1500°F. If this is not essential, it can be used up to 1800°F.

Powdered *diatomaceous silica* mixed with asbestos fiber is the base for a cement used to point up and fill cracks on the corresponding type of molded insulation. This cement is not

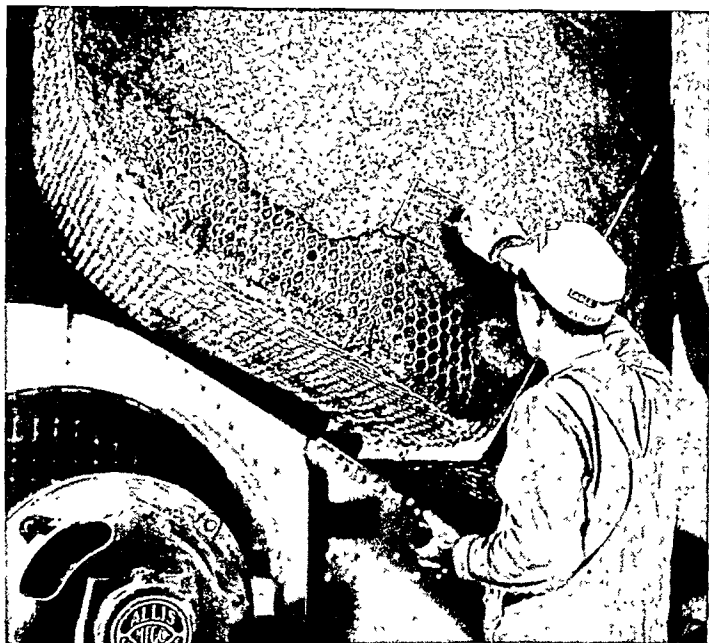


FIG. 114.—Showing the application of rock-wool insulating cement to form a monolithic covering. (Courtesy of the Eagle-Picher Co.)

generally used for hard finishing or for insulating large areas. It can be used up to 1600°F., or, if the diatomaceous silica has been calcined, up to 1900°F.

*Weatherproof* cements made of asbestos fibers and asphalt are used for protecting the insulation on outdoor equipment. These cements are supplied in plastic form and can be used at temperatures up to 400°F. They are usually applied in a single coat  $\frac{1}{4}$  in. thick over a coat of insulating cement.

*Adhesive cements* for temporarily holding sheet, block or brick insulation in place until the outer binding support is applied are also made. Where the temperatures do not exceed  $500^{\circ}\text{F}$  and where wire reinforcement is impractical, these cements may be used for permanently bonding the blocks to the metal surface or to each other.

**59 Insulating Concrete**—Coarsely ground calcined diatomaceous silica is used both as an insulating filler, as described in Sec 57, and as an ingredient in making insulating concrete. Four parts of the diatomaceous silica are mixed with one part of Portland cement by volume and enough water is added to form a plastic mass. The resulting concrete can be cast into any desired shape or it can be applied to steel surfaces by means of a cement gun. It has a thermal conductivity less than one-third that of fire-clay brick and can be exposed directly to temperatures as high as  $1800^{\circ}\text{F}$ . It can be used for the construction of enameling ovens, furnace doors, dampers, baffles and other heated equipment subjected to comparatively low temperatures.

**60 Insulating Brick**—Insulating bricks can be used in the construction of high temperature equipment such as heat treating furnaces, boiler settings, soaking pits and kilns. Two types are available: one used purely as an insulator and requiring the protection of a firebrick facing and the other used as a combination insulator and refractory. The latter type can be used to replace firebrick in furnaces where the walls are not subjected to slag action or mechanical abrasion. This construction permits the use of thinner walls with lower heat storage capacities and thereby decreases both the time required for heating up and the heat lost in cooling down the furnaces. A typical installation using the first type of brick is shown in Fig. 4 in Chap. I.

Various materials are used to make insulating bricks. One kind of brick is cut from natural diatomaceous silica. Another kind is made from ground diatomaceous silica pressed into bricks and calcined. Other kinds consist of refractory clay as a base, with various fibrous or cellular materials added to reduce the thermal conductivity. Insulating bricks are available in all standard firebrick sizes and shapes.

**61 Method of Calculating Heat Loss through Insulation**—The rate of heat loss through insulation on flat surfaces can be calculated by the equation



$$q = \frac{kA(t_s - t_a)}{L + (k/h_t)}, \quad (59)$$

where  $q$  = the rate of heat loss, B.t.u. per hr.

$k$  = the thermal conductivity of the insulation at its average temperature (approximately the average of  $t_s$  and  $t_a$ ), B.t.u./(ft.)(hr.)(°F.).

$A$  = the area of the surface that is covered with insulation, sq. ft.

$t_s$  = the temperature of the surface that is covered with insulation, °F.

$t_a$  = the temperature of the air surrounding the insulation, °F.

$L$  = the thickness of the insulation, ft.

$h_t$  = the "combined coefficient" for convection and radiation from the surface of the insulation (see Table 59), B.t.u./(sq. ft.)(hr.)(°F.).

Although the combined coefficient  $h_t$  depends upon the temperature of the surface of the insulation, sufficiently accurate values for practical purposes can usually be obtained if  $h_t$  is evaluated at any reasonable, assumed surface temperature. If more accurate values are required, the exact surface temperature can be calculated by a trial-and-error solution, as shown in the following problem:

**Illustrative Problem 1.**—A flat, vertical surface at a temperature of 500°F. is covered with 2 in. of 85% Magnesia. Calculate the exact rate of heat loss per square foot of surface if the surrounding air temperature is 70°F.

**Solution.**—As a first approximation, assume the surface temperature of the insulation to be 100°F. Based on this assumption, the average temperature of the insulation is 300°F.; the thermal conductivity of 85% Magnesia at this average temperature is 0.043 B.t.u./(ft.)(hr.)(°F.); and the combined coefficient  $h_t$ , from Table 59, is 1.68 B.t.u./(sq. ft.)(hr.)(°F.). Hence, by Eq. (59),

$$\begin{aligned} q &= \frac{0.043 \times 1 \times (500 - 70)}{(2/12) + (0.043/1.68)} \\ &= 96.1 \text{ B.t.u./(sq. ft.)(hr.).} \end{aligned}$$

If greater accuracy is desired, a more accurate value of the temperature  $t_s'$  of the surface of the insulation can be calculated by the equation

$$q = h_t A (t_s' - t_a).$$

Thus,

$$96.1 = 1.68 \times 1 \times (t_s' - 70),$$

or

$$t_s' = 127^\circ\text{F}$$

If the calculations are repeated using this surface temperature,  $q = 97.5$  Btu/(sq ft)(hr) is obtained

TABLE 59—COMBINED COEFFICIENT  $h_c$  FOR CONVECTION AND RADIATION FROM FLAT OR CYLINDRICAL SURFACES IN A ROOM AT  $70^\circ\text{F}$

The following table is calculated by the equation

$$h_c = h + \epsilon h_r$$

where  $h_c$  = the "combined coefficient" for convection and radiation, Btu/(sq ft)(hr)( $^\circ\text{F}$ )

$h$  = the film coefficient of convection (obtained from Case 12-14 or 18 in Chap IV) Btu/(sq ft)(hr)( $^\circ\text{F}$ )

$\epsilon$  = the emissivity of the surface of the insulation (assumed = 0.90)

$h_r$  = the radiation coefficient [calculated by Eq (21) in Chap III] Btu/(sq ft)(hr)( $^\circ\text{F}$ )

Type of surface	Temperature of surface				
	100 $^\circ\text{F}$	150 $^\circ\text{F}$	200 $^\circ\text{F}$	250 $^\circ\text{F}$	300 $^\circ\text{F}$
	$h_c$ Btu/(sq ft)(hr)( $^\circ\text{F}$ )				
<i>Flat surfaces</i>					
Vertical	1.68	2.07	2.38	2.67	2.95
Horizontal facing upward	1.86	2.32	2.66	2.98	3.28
Horizontal facing downward	1.46	1.77	2.03	2.29	2.54
<i>Cylindrical surfaces</i>					
Vertical	1.68	2.07	2.38	2.67	2.95
Horizontal 2 in. o.d.	1.98	2.40	2.73	3.03	3.32
Horizontal 4 in. o.d.	1.82	2.20	2.51	2.79	3.06
Horizontal 6 in. o.d.	1.75	2.10	2.40	2.66	2.93
Horizontal 8 in. o.d.	1.69	2.03	2.32	2.58	2.85
Horizontal 10 in. o.d.	1.66	1.99	2.27	2.53	2.79
Horizontal 12 in. o.d.	1.63	1.95	2.23	2.48	2.74

The rate of heat loss through insulation on pipes or other cylindrical surfaces can be calculated by the equation

$$q = \frac{k2\pi l(t_i - t_o)}{2.3 \log_{10}(r_2/r_1) + (k/h_o r_2)} \quad (60)$$

where  $r_1$  = the inside radius of the insulation ft

$r_2$  = the outside radius of the insulation ft,

$l$  = the length of the pipe, ft

and the other symbols have the same meanings as for Eq (59)

An approximate value for the rate of heat loss through insulation can be calculated by Eq. (3) or (4) in Chap. I if the temperature of the outside surface of the insulation is assumed to be the same as the temperature of the surrounding air. This procedure may introduce a considerable error if the thickness of the insulation is small or the temperature difference ( $t_s - t_a$ ) is large.

**Illustrative Problem 2.**—A horizontal 3-in. nominal-diameter pipe is covered with 1 in. of corrugated asbestos insulation (four ply per in.). If the temperature of the pipe is 300°F. and that of the surrounding air is 70°F., calculate the heat loss from the pipe per foot of length.

*Solution.*—The actual outside diameter of the pipe is 3.500 in.; so the outside diameter of the insulation is 5.5 in. Assuming that the surface temperature of the insulation is 125°F., the average temperature of the insulation is 213°F.; the thermal conductivity is 0.060 B.t.u./(ft.)(hr.)(°F.); and the combined coefficient  $h_c$ , from Table 59, is 1.95 B.t.u./(sq. ft.)(hr.)(°F.). Hence, by Eq. (60),

$$q = \frac{0.060 \times 2 \times 3.14 \times 1 \times (300 - 70)}{2.3 \log_{10} (2.75/1.75) + (0.060 \times 12)/(1.95 \times 2.75)} \\ = 148 \text{ B.t.u.}/(\text{hr.})(\text{ft. of length}).$$

*Approximate Solution.*—If the surface temperature of the insulation is assumed to be the same as the temperature of the surrounding air, the heat loss can be calculated by Eq. (4) in Chap. I:

$$q = \frac{0.060 \times 2 \times 3.14 \times 1 \times (300 - 70)}{2.3 \log_{10} (2.75/1.75)} \\ = 192 \text{ B.t.u.}/(\text{hr.})(\text{ft. of length}).$$

This value is approximately 30 per cent too high.

Equation (59) can be derived as follows: Letting  $t_s'$  denote the temperature of the outside surface of the insulation, the rate of heat transfer through the insulation by conduction [Eq. (3) in Chap. I] is

$$q = \frac{kA(t_s - t_s')}{L}.$$

Assuming steady flow conditions, this rate is equal to the rate at which heat is lost from the surface of the insulation by convection [Eq. (23) in Chap. III] and by radiation [Eq. (10) in Chap. II]:

$$q = hA(t_s' - t_a) + h_r A(t_s' - t_a)e.$$

or

$$q = h_i A(t_s' - t_a).$$

Equation (59) is obtained by eliminating  $t_i'$  from these equations. Equation (60) can be derived from Eqs (4), (10), and (23) in a similar manner.

**62. Most Economical Thickness of Insulation.**—As the thickness of insulation on any surface is increased, the rate of heat loss from the surface is decreased, but the cost of the insulation is increased. The most economical thickness of insulation is

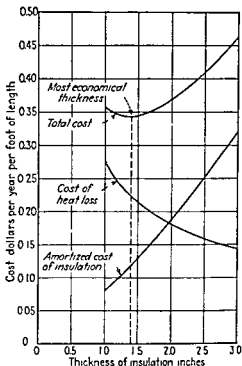


FIG. 115.—Determination of the most economical thickness of insulation (Data given in Illustrative Problem 3.)

that for which the sum of the yearly cost of the heat loss plus the yearly cost of the insulation is a minimum. The method of calculation is illustrated in the following problem.

**Illustrative Problem 3**—A 3-in. nominal diameter steam pipe is to be covered with 85% Magnesia. Determine the most economical thickness of insulation under the following conditions:

Temperature of the pipe = 400°F  
 Temperature of the surrounding air = 70°F

Time of operation per year	= 8 months
	= 5,760 hr.
Cost of the heat lost	= \$0.30 per 10 <sup>6</sup> B.t.u.
Cost of the insulation (applied):	
1-in. thickness	= \$0.53 per ft.
1½-in. thickness	= \$0.86 per ft.
2-in. thickness	= \$1.25 per ft.
3-in. thickness	= \$2.13 per ft.

Fraction of the cost of the insulation to be amortized each year = 0.15.

*Solution.*—The heat loss per hour per foot of length can be calculated for each thickness of insulation by Eq. (60). The yearly cost of the heat loss is equal to the product of this heat loss per hour times the hours of operation per year times the cost of the heat per B.t.u. The yearly cost of the insulation is equal to the product of the cost of the insulation (applied) times the fraction of this cost to be amortized each year. The results obtained from these calculations are shown in Fig. 115. For the data given, it would be most economical to cover the pipe with 1½ in. of insulation.

If the cost of the insulation per cubic foot is approximately constant for the different thicknesses, the most economical thickness can be calculated by the following equations developed by Patton:<sup>1</sup>

For *flat* surfaces,

$$L_0 = 12 \left[ \sqrt{\frac{nc'k(t_s - t_a)}{fc \cdot 10^6}} - \frac{k}{h_t} \right], \quad (61)$$

where  $L_0$  = the most economical thickness of insulation, in.

$n$  = the time that the equipment operates, hr. per year.

$c$  = the cost of the insulation (applied), dollars per cu. ft.

$c'$  = the cost of the heat lost, dollars per million B.t.u.

$f$  = the fraction of the cost of the insulation to be amortized each year.

$k$  = the thermal conductivity of the insulation, B.t.u./  
(ft.)(hr.)(°F.).

$h_t$  = the "combined coefficient" for convection and radiation from the surface of the insulation (see Table 59), B.t.u./(sq. ft.)(hr.)(°F.)

$t_s$  = the temperature of the surface to be insulated, °F.

$t_a$  = the temperature of the air surrounding the insulation, °F.

For *cylindrical* surfaces, the most economical ratio of outside to inside radius of insulation  $(r_2/r_1)_0$  can be determined from Table

<sup>1</sup> *Heating, Piping, and Air Conditioning*, vol. 4, p. 6, 1932.

60 after the quantity  $K_1$  has been calculated by the equation

$$K_1 = \frac{1}{r_1} \left[ \sqrt{\frac{nc'h(t_s - t_a)}{fc \cdot 10^4}} - \frac{h}{h_i} \right], \quad (62)$$

where  $r_1$  = the inside radius of the insulation, in

$r_2$  = the outside radius of the insulation, in

and the other symbols have the same meanings as for Eq. (61)

TABLE 60—MOST ECONOMICAL RATIO OF OUTSIDE TO INSIDE RADIUS OF INSULATION

$K_1$  calculated by Eq. (62)

$K_1$	$\left(\frac{r_2}{r_1}\right)_e$	$K_1$	$\left(\frac{r_2}{r_1}\right)_e$
0.00	1.000	0.24	2.799
0.02	1.218	0.26	2.916
0.04	1.407	0.28	3.031
0.06	1.578	0.30	3.144
0.08	1.737		
0.10	1.888	0.40	3.68
0.12	2.032	0.50	4.19
0.14	2.170	0.60	4.67
0.16	2.303	0.70	5.13
0.18	2.431	0.80	5.58
0.20	2.556	0.90	6.02
0.22	2.679	1.00	6.45

**Illustrative Problem 4**—Using the data given in Problem 3, determine the most economical thickness of insulation by means of Eq. (62) and Table 60.

**Solution**—The actual outside diameter of a 3-in. pipe is 3.500 in., so the inside radius of the insulation is 1.75 in. The average cost per cubic foot of the four thicknesses of insulation given in Problem 3 is approximately \$5.20. Assuming that the surface temperature of the insulation will be about 125°F, the average thermal conductivity of the insulation will be 0.042 Btu/(ft)(hr)(°F) and the combined coefficient  $h$  from Table 59, will be approximately 1.90 Btu/(q ft)(hr)(°F). Hence by Eq. (62),

$$K_1 = \frac{1}{1.75} \left[ \sqrt{\frac{5.760 \times 0.30 \times 0.042 \times (400 - 70)}{0.15 \times 5.20 \times 10^4}} - \frac{0.042}{1.90} \right] \\ = 0.0875$$

Therefore, from Table 60, the most economical ratio of outside to inside radius of insulation is

$$\left(\frac{r_2}{r_1}\right)_e = 1.794,$$

and

$$\begin{aligned} r_2 &= 1.75 \times 1.794 \\ &= 3.14 \text{ in.} \end{aligned}$$

Finally, the most economical thickness of insulation is  $3.14 - 1.75 = 1.39$  in.

Patton showed that, approximately,

$$K_1 = \frac{1}{12} \left( \frac{r_2}{r_1} \right)_0 \log_e \left( \frac{r_2}{r_1} \right)_0.$$

Table 60 was calculated from this equation.

**63. Thickness of Insulation Required to Prevent Freezing.**—The thickness of insulation required to prevent water freezing in pipes can be calculated by the equation

$$r_2 = r_1 \text{ antilog}_{10} \left[ \frac{2.73kl(t_w - t_a)}{G(t_w - 32)} \right], \quad (63)$$

where  $r_1$  = the inside radius of the insulation, in.

$r_2$  = the outside radius of the insulation, in.

$k$  = the thermal conductivity of the insulation, B.t.u./  
(ft.)(hr.)(°F.).

$l$  = the length of the pipe, ft.

$t_w$  = the temperature of the water entering the pipe, °F.

$t_a$  = the temperature of the air surrounding the pipe, °F.

$G$  = the rate at which the water flows, lb. per hr.

**Illustrative Problem 5.**—A  $1\frac{1}{2}$  in. water pipe is to be protected from freezing by a covering of hair felt. The water enters the pipe at 50°F., and the surrounding air is at -10°F. Calculate the thickness of insulation required if the water flows at a minimum rate of 5 gal. per hr.

**Solution.**—The actual outside diameter of a  $1\frac{1}{2}$ -in. pipe is 1.900 in.; so the inside radius of the insulation is 0.95 in. The thermal conductivity of hair felt is 0.022 B.t.u./ft.)(hr.)(°F.), and the weight rate of flow of the water is  $5 \times 8.35 = 41.8$  lb. per hr. By Eq. (63),

$$\begin{aligned} r_2 &= 0.95 \text{ antilog}_{10} \left[ \frac{2.73 \times 0.022 \times 100 \times (50 + 10)}{41.8 \times (50 - 32)} \right] \\ &= 2.86 \text{ in.} \end{aligned}$$

Hence, the insulation must be at least  $2.86 - 0.95 = 1.91$  in. thick.

If there is no flow of water through the pipe and the surrounding air temperature  $t_a$  remains below 32°F. indefinitely, freezing cannot be prevented regardless of the thickness of insulation used. The time required for water initially at

temperature  $t_w$  to be cooled to  $32^\circ\text{F}$  when there is no circulation can be calculated by the equation

$$\theta = \left( \frac{t_w - 32}{t_w - t_a} \right) \left( \frac{23}{l} + 0.04w \right) \frac{\log_{10} (r_2/r_1)}{k}, \quad (64)$$

where  $\theta$  = the time required for the water to cool to  $32^\circ\text{F}$ , hr  
 $l$  = the length of pipe containing 1 cu ft of water (see Table VI in the Appendix), ft  
 $w$  = the weight of the pipe (see Table VI in the Appendix), lb per ft of length  
 $t_w$  = the initial temperature of the water,  $^\circ\text{F}$ ,  
 and the other symbols have the same meanings as for Eq. (63)

**Illustrative Problem 6**—A  $1\frac{1}{2}$  in. wrought iron water pipe is covered with 2 in. of hair felt. If the surrounding air temperature is  $-10^\circ\text{F}$ , calculate the time required for water standing in the pipe to cool from an initial temperature of  $50^\circ\text{F}$  to a temperature of  $32^\circ\text{F}$ .

**Solution**—The actual outside diameter of  $1\frac{1}{2}$  in. pipe is 1.900 in., so the inside radius of the insulation is 0.95 in. and the outside radius is 2.95 in. One foot of the pipe weighs 2.717 lb. and 70.7 ft. contains 1 cu ft. of water. The thermal conductivity of hair felt is 0.022 Btu/(ft)(hr)( $^\circ\text{F}$ ). Hence the time  $\theta$  required for the water to cool to the freezing point is by Eq. (64)

$$\begin{aligned} \theta &= \left( \frac{50 - 32}{50 - 10} \right) \left( \frac{23}{70.7} + 0.04 \times 2.717 \right) \frac{\log_{10} (2.95/0.95)}{0.022} \\ &= 2.91 \text{ hr} \end{aligned}$$

Equation (63) can be derived by equating the rate at which the water gives up heat in cooling from  $t_w$  to  $32^\circ\text{F}$  to the rate at which heat is transferred by conduction through the insulation. In determining the latter the surface temperature of the insulation is assumed to be equal to  $t_a$  and the change in temperature of the water is neglected. Each of these assumptions results in an increased, and therefore more conservative, value for  $r_2$ .

Equation (64) can be derived by dividing the amount of heat given up by the water and by the metal pipe in cooling from  $t_w$  to  $32^\circ\text{F}$  by the rate at which heat is transferred by conduction through the insulation the same assumptions being made as for Eq. (63).

**64 Thickness of Insulation Required to Prevent Sweating**—The ratio of outside to inside radius of insulation ( $r_2/r_1$ ) required to prevent sweating on pipes can be determined from Table 61 after the quantity  $K_2$  has been calculated by the equation



$$K_2 = \frac{8k}{r_1} \left( \frac{t_d - t_w}{t_a - t_d} \right), \quad (65)$$

where  $r_1$  = the inside radius of the insulation, in.

$r_2$  = the outside radius of the insulation, in.

$k$  = the thermal conductivity of the insulation, B.t.u / (ft.)(hr.)(°F.).

$t_a$  = the dry-bulb temperature of the air surrounding the insulation, °F.

$t_d$  = the dew-point temperature of the air surrounding the insulation, °F.

$t_w$  = the temperature of the fluid inside the pipe, °F.

TABLE 61 —RATIO OF OUTSIDE TO INSIDE RADIUS OF INSULATION REQUIRED TO PREVENT SWEATING ON PIPES  
 $K_2$  calculated by Eq. (65)

$K_2$	$\left(\frac{r_2}{r_1}\right)$	$K_2$	$\left(\frac{r_2}{r_1}\right)$
0 00	1 000	1 20	2 741
0 10	1 210	1 30	2 854
0 20	1 393	1 40	2 965
0 30	1 559	1 50	3.075
0 40	1 713		
0 50	1 858	2 00	3 60
0 60	1 997	2 50	4 09
0 70	2 130	3 00	4 56
0 80	2 259	3 50	5 01
0 90	2 384	4 00	5 44
1 00	2 506	4 50	5 86
1 10	2 625	5 00	6 27

**Illustrative Problem 7.**—A 2-in. pipe carrying water at 50°F is to be covered with sufficient wool felt to prevent sweating. Calculate the thickness of insulation required if the temperature of the surrounding air is 90°F. and the relative humidity is 70 per cent.

**Solution.**—The actual outside diameter of a 2-in. pipe is 2.375 in.; so the inside radius of the insulation will be 1.188 in. The dew-point temperature corresponding to a dry-bulb temperature of 90°F. and a relative humidity of 70 per cent is 79°F. (determined from any psychrometric chart). The average temperature of the insulation will be 65°F., and its thermal conductivity at this temperature will be 0.024 B.t.u / (ft.)(hr.)(°F.). By Eq. (65),

$$K_2 = \frac{8 \times 0.024}{1.188} \left( \frac{79 - 50}{90 - 79} \right) \\ = 0.426.$$

Therefore from Table 61, the ratio of the outside to the inside radius of the insulation must be

$$\frac{r_2}{r_1} = 1.751,$$

or

$$\begin{aligned} r_2 &= 1.751 \times 1.188 \\ &= 2.08 \text{ in} \end{aligned}$$

Hence, the thickness of the insulation must be at least

$$2.08 - 1.188 = 0.89 \text{ in}$$

Equation (65) can be derived as follows. The minimum thickness of insulation required to prevent sweating is that thickness for which the surface temperature is exactly equal to the dew-point temperature of the surrounding air. This thickness can be found by equating the rate of heat transfer through the insulation by conduction to the rate of heat transfer from the surface of the insulation to the surrounding air by convection and by radiation. Thus, per foot of pipe,

$$\frac{k2\pi(t_d - t_w)}{2.3 \log_{10}(r_2/r_1)} = h_i \left( \frac{2\pi r_2}{12} \right) (t_a - t_d),$$

where  $h_i$  is the "combined coefficient" for convection and radiation (see Table 59). If  $h_i$  is assumed equal to 0.65 Btu/(sq ft)(hr)(°F) [the smaller this value the more conservative the result] this equation reduces to

$$\left( \frac{r_2}{r_1} \right) \log_{10} \left( \frac{r_2}{r_1} \right) = \frac{8k}{r_1} \left( \frac{t_d - t_w}{t_a - t_d} \right)$$

Equation (65) is obtained by letting  $K_2 = (r_2/r_1) \log_{10}(r_2/r_1)$ . Table 61 is based on the latter equation defining  $K_2$ .

# APPENDIX

TABLE I.—THERMAL CONDUCTIVITIES OF METALS\*  
Linear interpolation between temperatures is permissible

Metal	Temperature, °F	Thermal conductivity $k$ B t u / (ft ) (hr ) (°F )
Aluminum	0	116
	200	119
	400	124
	600	134
	900	153
Antimony	32	10 6
	212	9 7
Bismuth	64	4 7
	212	3 9
Brass (70 30)	0	55
	800	68
Bronze	68	33 6
	210	41 0
Cadmium	64	53 7
	212	52 2
Cast iron	32	29
	212	28
Constantan	64	13 1
	212	15 5
Copper (pure)	0	225
	200	218
	1200	202
Gold	64	169
	212	170
Iron (pure)	64	39 0
	212	36 6
Lead	0	21
	400	19
Magnesium	32	92
	212	92
Mercury	32	3 6
	122	4 6
Nickel	0	35
	600	32
Nickel silver	32	16 9
	212	21 5
Platinum	64	40 2
	212	41 9
Silver	32	242
	212	238
Steel (mild) cold rolled	200	26
	1100	21
Tin	32	37
	392	34
Tungsten	68	92 5
Wrought iron	64	34 9
	212	34 6
Zinc	0	65
	400	61
	800	54

\* Compiled from "International Critical Tables," Marks "Mechanical Engineers Handbook," and other sources

TABLE II—THERMAL CONDUCTIVITIES OF INSULATING MATERIALS\*  
 Linear interpolation between temperatures is permissible. Section numbers  
 refer to the section in which the material is described.

Material	Apparent density $\rho$ lb per cu ft.	Temper- ature °F	Thermal conductivity $k$ B.t.u. / (ft.) (hr) (°F)
<i>Pipe covering and blocks</i>			
85% Magnesia (Secs. 50b and 53a)	17	100	0.039
	17	500	0.047
Diatomaceous silica, asbestos and bonding material (Secs. 50c and 53a)	23	100	0.045
	23	1000	0.063
Laminated asbestos felt (40 ply per in.) (Secs. 50d and 53a)	30	100	0.033
	30	500	0.048
Laminated asbestos felt (20 ply per in.)		100	0.045
		500	0.065
Felted asbestos (Secs. 50e and 53a)	36	100	0.097
	36	200	0.110
	36	300	0.118
	36	800	0.139
	44	-300	0.100
	44	-50	0.132
	44	32	0.135
Felted rock or glass wool (Secs. 50f and 50g)	16	100	0.030
	16	600	0.057
Corrugated asbestos (4 ply per in.) (Sec. 50j)	12	100	0.050
	12	300	0.068
Wool felt (Secs. 50k and 52e)	8.5	32	0.072
	8.5	200	0.033
	20.6	80	0.03
Cork, molded (Secs. 52b and 54a)	7.5	32	0.021
	7.5	122	0.074
	12.5	32	0.030
	12.5	122	0.031
Rock wool and binder (Secs. 52c and 54z)	15	32	0.025
	15	60	0.077
Hair felt (Sec. 52d)	9	70	0.022
	17	70	0.022
Crumpled aluminum foil ( $\frac{3}{8}$ in. air spaces) (Sec. 50l)	0.2	100	0.025
	0.2	350	0.038
<i>Sheets and blankets</i>			
Asbestos millboard (Sec. 55)	55	124	0.096
Asbestos sheets (Sec. 55)	36	100	0.097
	36	200	0.110
	36	300	0.118
	36	800	0.130
	44	32	0.135
Corrugated asbestos (4 ply per in.) (Sec. 55)	12	100	0.050
	12	300	0.068
Hair felt (Sec. 55)	9	70	0.022
	17	70	0.022
Hair and asbestos fiber, felted	7.8	70	0.023

\* Compiled from "International Critical Tables," Marks, Mechanical Engineers' Handbook, and other sources.

TABLE II—THERMAL CONDUCTIVITIES OF INSULATING MATERIALS \*—(Continued)

Material	Apparent density $\rho$ lb per cu ft	Temperature, °F	Thermal conductivity $k$ , B t u / (ft ) (hr ) (°F )
<i>Sheets and blankets (Continued)</i>			
Wool felt	8 5	32	0 022
	8 5	200	0 023
	20 6	86	0 03
Aluminum foil (2½-in spacing) (Sec 55)	0 2	100	0 025
	0 2	350	0 038
Felted rock or glass wool (Sec 56)	16	100	0 030
	16	600	0 057
<i>Fillers*</i>			
Diatomaceous silica, powdered (Sec 57)	10 6	70	0 026
	17 2	100	0 032
	17 2	1600	0 074
	20 0	100	0 036
	20 0	1600	0 082
Rock or glass wool loose or granular (Sec 57)	6	70	0 022
	18	70	0 024
Asbestos, loose (Sec 57)	29 3	-300	0 054
	29 3	32	0 089
	29 3	210	0 093
Cork, ground (Sec 57)	10 0	32	0 021
	10 0	200	0 032
Cork, regranulated (Sec 57)	8 1	86	0 026
Magnesia powdered	49 7	117	0 35
Gypsum, powdered	26	70	0 047
	34	70	0 047
Wool (animal), loose	6 9	86	0 021
Wood fiber, chemically treated	2 2	70	0 023
<i>Insulating cements</i>			
85% Magnesia (Sec 58)			0 6
Asbestos (Sec 58)			1 2
Asbestos and rock wool (Sec 58)			0 9
<i>Insulating concrete</i>			
Four parts diatomaceous silica and one part cement (Sec 59).	61 8	400	0 16
	61 8	1600	0 23
<i>Insulating brick</i>			
Diatomaceous silica, natural perpendicular to strata (Sec 60)	27 7	400	0 051
	27 7	1600	0 077
Diatomaceous silica, natural, parallel to strata (Sec 60)	27 7	400	0 081
	27 7	1600	0 106
Diatomaceous silica, calcined (Sec 60)	38	400	0 14
	38	1600	0 18
Diatomaceous silica and fire clay, calcined (Sec 60)	42 3	400	0 14
	42 3	1600	0 19
Kaolin insulating brick	27	932	0 15
	27	2100	0 26

\* Compiled from "International Critical Tables" Marks "Mechanical Engineers' Handbook," and other sources

TABLE IV—EMISSIVITY OF SURFACES \*—(Continued)

Surface	Temperature, † °F	Emissivity†
<i>Nickel alloy (Continued)</i>		
Same after 24 hr at 980°F	420-980	0.62-0.73
NCT-3 (20% Ni, 25Cr, brown spotted, oxidized from service)	420-980	0.90-0.97
NCT-6 (60Ni 12Cr smooth black firm adhesive oxide coat from service)	520-1045	0.89-0.82
<i>Platinum</i>		
Polished plate	440-2500	0.054-0.17
Platinum filament	80-2240	0.036-0.192
Silver, pure polished	440-1160	0.02-0.032
Tantalum filament	2420-5430	0.194-0.31
Tin bright	76	0.043-0.064
Tungsten aged filament	80-6000	{ 0.032-0.35 0.018
<i>Zinc</i>		
Commercial polish	440-620	0.045-0.053
Oxidized at 750°F	750	0.11
Galvanized iron fairly bright	82	0.23
Galvanized iron gray oxidized	75	0.28
Refractories Building Materials Paints Miscellaneous		
<i>Aluminum paints (vary with amount of lacquer body and age)</i>	212	0.27-0.67
<i>Asbestos</i>	100-700	0.93-0.95
<i>Carbon</i>		
Candle soot lampblack—waterglass	70-700	0.95±0.01
<i>Oil</i>		
Lubricating layer 0.01 in thick	68	0.82
Linseed one and two coats on Al foil	68	0.56 and 0.57
<i>Rubber soft gray reclaimed</i>	76	0.839
<i>Miscellaneous I</i> shiny black lacquer planed oak white enamel serpentine gypsum white enamel paint roofing paper lime plaster black matte shellac	70	0.87-0.91
<i>Miscellaneous II</i> glazed porcelain white paper fused quartz polished marble rough red brick smooth glass hard glossy rubber flat black lacquer water electrographite	70	0.92-0.96

\* Prepared by H. C. Hottel

† When two temperatures and two emissivities are given, they correspond first to first and second to second and linear interpolation is permissible.

TABLE V.—DIMENSIONS OF FREQUENTLY USED SIZES OF TUBING

Outside tube, diameter, in	Wall thickness		Inside tube diameter in	Surface per ft of length		Length of tube per sq ft of surface		Cross sectional area	
	Birmingham Wire Gauge (B W G )	In		Outside surface sq ft per ft of length	Inside surface, sq ft per ft of length	Out-side, ft	In-side ft	Out-side, sq in	In-side, sq in
$\frac{1}{2}$	20	0 035	0 430	0 13090	0 11257	7 6394	8 8831	0 19635	0 14522
	19	0 042	0 416		0 10891		9 1820		0 13592
$\frac{3}{8}$	18	0 049	0 527	0 16362	0 13797	6 1116	7 2480	0 30680	0 21813
	16	0 065	0 495		0 12959		7 7166		0 19244
	14	0 083	0 459		0 12017		8 3218		0 16547
$\frac{3}{4}$	18	0 049	0 652	0 19635	0 17069	5 0930	5 8585	0 44179	0 33388
	16	0 065	0 620		0 16232		6 1608		0 30191
	14	0 083	0 584		0 15289		6 5406		0 26787
$\frac{7}{8}$	18	0 049	0 777	0 22907	0 20342	4 3654	4 9160	0 60132	0 47417
	16	0 065	0 745		0 19504		5 1271		0 43592
	14	0 083	0 709		0 18562		5 3875		0 39481
1	18	0 049	0 902	0 26180	0 23614	3 8197	4 2347	0 78540	0 63900
	16	0 065	0 870		0 22777		4 3905		0 59447
	14	0 083	0 834		0 21834		4 5800		0 54629
$1\frac{1}{4}$	18	0 049	1 152	0 32725	0 30159	3 0558	3 3157	1 2272	1 04231
	16	0 065	1 120		0 29322		3 4105		0 98520
	14	0 083	1 084		0 28379		3 5237		0 92289
$1\frac{1}{2}$	16	0 065	1 370	0 39270	0 35867	2 5405	2 7881	1 7672	1 4741
	14	0 083	1 334		0 34924		2 8634		1 3977
2	14	0 083	1 834	0 52360	0 48014	1 9099	2 0827	3 1416	2 6417
	12	0 109	1 782		0 46653		2 1435		2 4941

TABLE VI — STANDARD DIMENSIONS FOR STANDARD-WEIGHT WROUGHT-IRON PIPE\*

Nominal size, in.	Actual diameters		Nominal thickness, in.	Circumference		Transverse areas		Length of pipe per sq ft.		Nominal weight, lb./ft.	Number of threads per inch of screw
	External, in.	Approximate internal, in.		External, in.	Internal, in.	External, sq in.	Internal, sq in.	External surface, ft.	Internal surface, ft.	Plain ends	Threaded and coupled
3/4	0 405	0 269	0 068	1 272	0 845	0 129	0 037	9 431	14 199	0 244	0 245
1	0 540	0 364	0 088	1 696	1 144	0 209	0 104	7 073	10 493	0 424	0 425
1 1/4	0 675	0 453	0 091	2 121	1 419	0 345	0 191	5 638	7 747	0 567	0 568
1 1/2	0 810	0 522	0 109	2 529	1 654	0 534	0 304	4 547	6 141	0 806	0 807
2	1 040	0 684	0 113	3 279	2 269	0 866	0 533	3 637	4 635	1 130	1 131
2 1/2	1 315	0 949	0 135	4 131	3 296	1 355	0 864	2 904	3 641	1 672	1 673
3	1 600	1 280	0 140	5 215	4 315	2 164	1 495	2 201	2 767	2 272	2 273
3 1/2	1 900	1 610	0 145	5 969	5 048	2 835	2 036	1 608	1 847	3 042	3 043
4	2 275	2 067	0 154	7 461	6 494	4 430	3 355	1 204	1 547	3 703	3 704
4 1/2	2 675	2 469	0 203	8 632	7 737	6 497	4 788	1 091	1 245	4 372	4 373
5	3 000	3 068	0 216	10 006	9 638	9 621	7 393	0 954	1 076	5 100	5 101
5 1/2	3 400	3 548	0 226	12 568	11 146	12 568	9 856	0 846	0 943	5 819	5 820
6	4 000	4 028	0 237	14 137	12 648	15 904	12 730	0 763	0 847	6 602	6 603
6 1/2	4 500	4 503	0 247	15 768	14 146	19 635	15 047	0 686	0 756	7 412	7 413
7	5 263	5 047	0 268	17 477	15 846	24 306	20 066	0 576	0 629	8 281	8 282
7 1/2	6 025	6 063	0 280	20 813	19 581	34 472	28 891	0 500	0 543	9 150	9 151
8	7 075	7 075	0 301	23 945	22 063	41 664	33 755	0 442	0 473	10 020	10 021
8 1/2	8 025	8 071	0 327	27 096	25 348	51 416	41 181	0 386	0 427	10 900	10 901
9	9 025	9 081	0 352	30 238	28 073	58 426	46 027	0 336	0 374	11 781	11 782
9 1/2	10 025	10 124	0 379	33 772	31 019	72 760	62 786	0 306	0 351	12 662	12 663
10	11 025	11 125	0 403	37 772	34 479	80 763	80 691	0 286	0 331	13 543	13 544
10 1/2	12 025	12 125	0 428	41 813	38 146	90 763	90 691	0 266	0 311	14 424	14 425
11	13 025	13 125	0 453	45 813	41 813	100 763	100 691	0 246	0 291	15 305	15 306
12	14 025	14 125	0 478	50 000	46 000	110 763	110 691	0 226	0 271	16 186	16 187
12 1/2	15 025	15 125	0 503	54 000	50 000	120 763	120 691	0 206	0 251	17 067	17 068

\* From W. H. McAdams "Heat Transmission" McGraw-Hill Book Company, Inc. 1933



## INDEX

### A

Absorption of radiation, 30  
 Absorptivity, of black body, 33  
     definition, 32  
     and emissivity, equivalence of, 33  
     of gray surface, 34  
 Acetic acid, film coefficients, 64  
     pressure drop, 122-134  
 Acetone, film coefficients, 64  
     pressure drop, 122-134  
 Acetylene, film coefficients, 64  
     pressure drop, 128-135  
 Adhesive cement, 183, 186, 198  
 Air, film coefficients, 64  
     pressure drop, 128-135  
 Alloys, thermal conductivity of, 209  
 Aluminum foil insulation, 171, 191,  
     192  
 Ammonia, film coefficients, 64  
     pressure drop, 122-134  
 Amyl acetate, film coefficients, 64  
     pressure drop, 122-134  
 Amyl alcohol (iso), film coefficients,  
     64  
     pressure drop, 122-134  
 Aniline, film coefficients, 64  
     pressure drop, 122-134  
 Annular spaces, equivalent inside  
     diameter, 94  
     film coefficients, 59-94  
     pressure drop, 117, 134  
 Application of insulation, method  
     (see kind of insulation)  
 Area, effective for radiation, 24  
     of heat-transfer surface, 52  
     for standard pipe (table), 216  
     of tubing (table), 215  
 Asbestos, 162  
     blankets, quilted, 168, 195  
     cement, 172-175, 196

Asbestos, corrugated, 170, 190  
     and diatomaceous silica, 165, 182,  
         197  
     felted, 166, 183, 190  
     laminated, 166, 182  
     and magnesium carbonate, 164  
         182, 196  
     millboard, 189  
     paper, 189  
 Average temperature, for evaluating  
     film coefficient, 53  
 Average temperature difference, 46-  
     52

### B

Baffles, 136, 139-141  
     disk-and-ring, 140  
     helical, 141  
     impact, 141  
     longitudinal, 139, 141  
     method of support, 139, 140  
     orifice-type, 140  
     segmental, 140  
     transverse, 136, 139-141  
     vane-type, 140  
 Base value, of film coefficient, defi-  
     nition, 65  
     of pressure drop, definition, 118  
 Basic insulating materials, 162  
 Basket-type evaporator, 147, 148  
 Benzene, film coefficients, 64  
     pressure drop, 122-134  
 Black body, definition, 23, 33  
 Blankets, glass wool, 194  
     insulating, 164  
     quilted asbestos, 168, 195  
     rock wool, 194  
 Block insulation, 164, 182-189  
 Boiler walls, method of insulating,  
     185

- Boiling liquids (*see* Evaporation)
- Boltzmann, 34
- Bowed tubes, 138, 144
- Brick, insulating, 198
- Brine ( $\text{CaCl}_2$ ), film coefficients, 64
  - pressure drop, 122-134
- Butane, film coefficients, 64
  - pressure drop, 128-135
- Butyl alcohol (n), film coefficients, 64
  - pressure drop, 122-134

## C

- Calculation of charts and tables, 56-63, 113-118
  - of heat loss through insulation, 198-202
- Carbon dioxide, film coefficients, 64
  - pressure drop, 128-135
- Carbon disulfide, film coefficients, 64
  - pressure drop, 122-134
- Carbon monoxide, film coefficients, 64
  - pressure drop, 128-135
- Carbon tetrachloride, film coefficients, 64
  - pressure drop, 122-134
- Cement, adhesive, 183, 186, 198
  - asbestos, 172-175, 196
  - expanded vermiculite, 196
  - glass wool, 196
  - rock wool, 196
  - weatherproof, 197
- Charts and tables, for calculating
  - pressure drop, 122-135
  - calculation of, 113-118
  - illustrative problems, 118-121
  - use of, 118
  - for film coefficients, 80-112
    - calculation of, 56-63
    - illustrative problems, 66-79
    - index to, 64
    - use of, 63-66
- Chlorine, film coefficients, 64
  - pressure drop, 128-135
- Chlorobenzene, film coefficients, 64
  - pressure drop, 122-134
- Chloroform, film coefficients, 64
  - pressure drop, 122-134
- Cleaning, of tubes, 137, 160
- Cleaning fluids, 161
- Cleaning lanes, 158
- Coefficient of heat transfer, film (*see* Film coefficient)
  - over-all (*see* Over-all coefficient)
  - radiation (*see* Radiation, coefficient of heat transfer)
- Coiled tube evaporator, 152, 153
- Coils, film coefficients for, 60, 61, 104
  - pipe, 154
- Combination insulation, 12, 165
- Combined coefficient (table), 200
- Concentric tubes, 144
- Concrete, insulating, 198
- Condensation, discussion of, 39, 40
  - dropwise, 39
  - film coefficients for, 62, 106, 108
    - outline of, 64
  - filmwise, 39
  - in insulation, 176, 180
  - on pipes, prevention of, 206-208
- Condenser, definition, 136
- Condensers, mean temperature difference in, 46, 47
- Conduction, 3-22
  - definition, 1
  - fundamental equation for, 5
  - resistance, definition, 13
  - steady, calculation of, 6-22
    - through bodies of any shape, 10
    - through composite bodies, 12-17
      - through cylindrical bodies, 7
      - through flat bodies, 6
      - in fluids, 5
    - illustrative problems, 7, 8, 10, 13
    - through rectangular bodies, 11
    - through spherical bodies, 9
  - definition, 5
  - unsteady, calculation of, through
    - flat bodies, 17-22
    - illustrative problem, 19
  - definition, 5

- Conductivity, thermal (*see* Thermal conductivity)
- Contact, effect on conduction, 16
- Contraction, effect on pressure (*see* Entrance to tubes)
- Contraction coefficient, 115
- Convection, 37-53  
     definition, 1, 37  
     and radiation, combined coefficient for (table), 200
- Convection coefficients, 54-112  
     charts and tables, 80-112  
     index to, 64  
     illustrative problems, 66-79  
     (*see also* Film coefficient; Overall coefficient)
- Cooler, definition, 136
- Cooling, effect of, on film coefficient, 39
- Cork, 163, 176, 177, 187-189  
     granulated, 177, 188
- Cork insulation, 176, 187-189
- Corrugated asbestos insulation, 170, 190
- Corrugation in shell, 144
- Counterflow, definition, 47
- Crossflow heat exchanger, mean temperature difference in, 47, 51
- Cutters, scale, 159
- D**
- Density, of gases, equation for, 130  
     of liquids (table), 122
- Density changes, effect of, on pressure drop, 113, 117, 134
- Design of equipment, 52, 53  
     illustrative problems, 66-79, 118-121
- Diameter correction factor, 65, 118
- Diameters, of tubing (table), 215  
     of wrought-iron pipe (table), 216
- Diatomaceous earth (*see* Diatomaceous silica)
- Diatomaceous silica, 163  
     with asbestos fiber, 165, 182, 197  
     bricks, 198  
     powdered, 195
- Dimensional analysis, 41, 54, 56  
     illustration, 54-56
- Disk-and-ring baffles, 140
- Double-pipe heat exchanger, 144-146  
     illustrative problem, 73
- Downtake, 146, 148-150
- Drifting, 159
- Dropwise condensation, 39
- Drums, method of insulating, 184, 192, 194
- Dulong, 34
- E**
- Economic thickness of insulation, 203-205
- Eighty-five per cent (85%) magnesia, 164, 182, 196
- Emission of radiation, 30
- Emissivity, and absorptivity, equivalence of, 33  
     definition of, 24, 31  
     effect of temperature on, 24  
     of gray surface, 24, 33  
     hemispherical, definition, 33  
     monochromatic, definition, 31  
     of surfaces (table), 213, 214  
     table of, 213  
     total, definition, 33
- Emissivity factor, definition, 26  
     table of, 26
- Enlargement, effect on pressure (*see* Outlet from tubes)
- Entrance to tubes, pressure drop at, 113, 115, 116, 126, 130
- Equivalent diameter, 94, 95
- Ethane, film coefficients, 64  
     pressure drop, 128-135
- Ethyl acetate, film coefficients, 64  
     pressure drop, 122-134
- Ethyl alcohol, film coefficients, 64  
     pressure drop, 122-134
- Ethyl bromide, film coefficients, 64  
     pressure drop, 122-134
- Ethyl chloride, film coefficients, 64  
     pressure drop, 122-134
- Ethyl ether, film coefficients, 64  
     pressure drop, 122-134

- Ethyl iodide, film coefficients, 64
  - pressure drop, 122-134
- Ethylene, film coefficients, 64
  - pressure drop, 128-135
- Ethylene glycol, film coefficients, 64
  - pressure drop, 122-134
- Evaporation, of boiling liquids, discussion of, 40, 41
  - film coefficients, 63, 110
  - outline of, 64
- Evaporator, definition, 136
  - illustrative problem, 76-78
- Evaporators, 146-152
  - basket type, 147, 148
  - coiled-tube, 152, 153
  - forced circulation, 151, 152
  - horizontal tube, 150
  - inclined tube, 149
  - Robert type, 146
  - standard vertical short tube, 146, 147
  - vertical long tube, 148
  - Wellner-Jelinek, 150
- Expanding tubes, 159
- Expansion, provision for, 137, 142-144

## F

- Fanning equation*, 114
- Felted asbestos insulation, 166, 183, 190
- Ferrule, 144
- Fillers, insulating, 195
- Film coefficient, definition, 38
  - factors affecting, 38-41
  - and over all coefficient relation between, 42-45
- Film coefficients bases for charts and tables* 56-63
  - charts and tables of, 80-112
  - fluid temperature used in evaluating, 53
  - illustrative problems, 66-79
  - index to charts and tables 64
  - use of charts and tables, 63-66
- Film temperature, definition, 63, 118
- Filmwise condensation* 39
- Finishing of insulation (*see* kind of insulation)
- Finned tubes, 138, 145
- Flanges, insulation on, 173-176
- Flared tubes, 159
- Flat surfaces, heat loss through insulation on, 198
  - method of insulating, 183 187, 192, 194
- Flexible flanges in shell, 144
- Floating-head, 142, 143
- Forced circulation evaporator, 151, 152
- Forced convection, definition, 1, 37
- Fourier's law, 5
- Freezing, prevention of, 203
- Friction (*see* Pressure drop)
- Friction factor, 114
- Full floating head, 142

## G

- Gas densities, equation for, 130
- Gases, film coefficients for, 57-61, 84-104
  - annular spaces, turbulent flow, 59, 94
  - flat plates, natural convection, 60, 61, 102 104
  - inside coils, turbulent flow, 61, 104
  - inside tubes natural convection, 60, 102
    - turbulent flow, 57, 84, 86
  - outline of, 64
  - outside coils, forced convection, 61 104
    - natural convection, 61, 104
  - outside single tubes, natural convection, 60, 98, 102
    - normal flow, 58, 90, 92
  - outside tube bundles, natural convection, 60, 102
    - normal flow, 58, 94
    - parallel flow, 59, 94
  - pressure drop, 116-118, 128-135
    - in annular spaces, 117, 134
    - caused by changes in density, 117, 134

Gases, pressure drop, at entrance to tubes, 116, 130  
 at outlet from tubes, 117, 132  
 inside smooth tubes, 116, 128  
 outside tube bundles, 117, 134  
 Generator air cooler, 153  
 Glass wool, granulated, 195  
 loose, 195  
 manufacture of, 163  
 Glass wool blankets, 194  
 Glass wool cement, 196  
 Glass wool insulation, 168  
 Glycerol, film coefficients, 64  
 pressure drop, 122-134  
 Granulated cork, 177, 188  
 Granulated glass wool, 195  
 Granulated rock wool, 188, 195  
 Gray surface, absorptivity of, 34  
 definition, 33  
 emissivity of, 24, 33  
 Grooves, in tube-sheet holes, 159

## H

$h$ , film coefficient of heat transfer, definition, 38  
 Hair felt, 179-181, 191  
 Hairpin tubes, 138  
 Half sections, insulation, 163  
 Heat exchanger, definition, 136  
 double-pipe, 144-146  
 Heat exchangers, mean temperature difference in, 46-52  
 Heat loss, through insulation, 198-202  
 Heat regenerators, 158  
 Heat transfer, coefficient of (*see* Film coefficients; Over-all coefficients)  
 Heat transfer equipment, design of, 52, 53  
 types of, 136-161  
 Heater, definition, 136  
 hot-water, 155-157  
 tank suction, 146, 147  
 Heating, effect of, on film coefficient, 39  
 Helical baffles, 141

Helical coils (photograph), 155  
 Helium, film coefficients, 64  
 pressure drop, 128-135  
 Heptane, film coefficients, 64  
 pressure drop, 122-134  
 Hexane, film coefficients, 64  
 pressure drop, 122-134  
 Horizontal-tube evaporator, 150  
 Hot-water heaters, 155-157  
 Hydrogen, film coefficients, 64  
 pressure drop, 128-135  
 Hydrogen sulfide, film coefficients, 64  
 pressure drop, 128-135

## I

Impact baffles, 141  
 Inclined-tube evaporator, 149  
 Index to charts and tables, film coefficients, 64  
 Individual coefficient (*see* Film coefficient)  
 Industrial heat-transfer equipment, 136-161  
 Infusorial earth (*see* Diatomaceous silica)  
 Installation of tubes, 159  
 Instantaneous heaters, hot-water, 155, 157  
 Insulating brick, 198  
 Insulating concrete, 198  
 Insulation, 162-208  
 aluminum foil, 171, 191, 192  
 asbestos cement, 172-175, 196  
 basic materials, 162  
 blankets, 164  
 blocks, 164, 182-189  
 brick, 198  
 calculation of heat loss, 198-202  
 cements, 196  
 for cold pipes, 176-182  
 combination, 12, 165  
 concrete, 198  
 condensation in, 176, 180  
 cork, 176, 187-189  
 corrugated asbestos, 170, 190  
 diatomaceous silica and asbestos fiber, 165, 182, 197

Insulation, most economical thickness, 202-205  
 felted asbestos, 166, 183, 190  
 fillers, 195  
 finishing of (*see* kind of insulation)  
 glass wool, 168  
 glass wool blankets, 194  
 granulated, 195  
 hair felt, 179-181, 191  
 half sections, 163  
 for hot pipes, 163-172  
 laminated asbestos-felt, 166, 182  
 method of applying (*see* kind of insulation)  
 molded, 164, 166, 176, 177, 182  
 powdered diatomaceous silica, 195  
 properties required, 162  
 quilted asbestos blankets, 168, 195  
 rock wool, 167, 177-179  
 rock wool blankets, 194  
 sheets, 189-192  
 tape, 164, 170  
 thermal conductivity of (table), 210, 211  
 thickness of to prevent freezing, 205  
     to prevent sweating 206-208  
 valves and fittings, 172-179, 181  
 whole sections 163  
 wool felt 171-181  
 Intensity of radiation (*see* Radiation, intensity of)  
 Iron pipe, dimensions of (table), 216

## J

Jacketed vessels, 157

## K

Kirchhoff's law, 32  
 Knockers, 161

## L

Lags, definition, 186  
     (*See also* Block insulation)  
 Lambert's cosine principle, 32

Laminar flow (*see* Streamline flow)  
 Laminated asbestos-felt insulation, 166, 182  
 Liquids, boiling (*see* Evaporation)  
     film coefficients for, 57-63, 80-111  
     annular spaces, turbulent flow, 59, 94  
     flat plates, natural convection, 60, 100  
     inside coils, turbulent flow, 60, 104  
     inside tubes, natural convection, 60, 100  
         streamline flow, 61, 62, 105  
         turbulent flow, 57, 80, 82  
     outline of, 64  
     outside coils, forced convection 61, 104  
         natural convection, 61, 104  
     outside single tubes, natural convection, 59, 60, 96, 100  
         normal flow, 58, 88  
     outside tube bundles, natural convection, 60, 100  
         normal flow, 58, 94  
         parallel flow, 59, 94  
 pressure drop, 114-118, 122-134  
     in annular spaces, 117, 134  
     caused by reversal in flow, 117, 134  
     at entrance to tubes, 115, 126  
     at outlet from tubes, 116, 127  
     inside smooth tubes, 114, 122  
     outside tube bundles, 117, 134  
 Logarithmic mean temperature difference, chart for determining, 48  
     definition, 47  
     equation for, 47  
 Long tube evaporator, 148  
 Longitudinal baffles, 139, 141

## M

Magnesia (80%), 164, 182, 196  
 Magnesium carbonate, 162  
     with asbestos fiber, 164, 182, 196

- Mean temperature difference (*see* Temperature difference)
- Metals, thermal conductivity of (table), 209
- Methane, film coefficients, 64  
pressure drop, 128-135
- Methyl alcohol, film coefficients, 64  
pressure drop, 122-134
- Methyl chloride, film coefficients, 64  
pressure drop, 128-135
- Millboard, asbestos, 189
- Molded insulation, 164, 166, 176, 177, 182
- Multipass, definition, 141
- Multipass heat exchanger, mean temperature difference in, 47, 49, 50
- N
- Natural convection, definition, 1, 37
- Nitric oxide, film coefficients, 64  
pressure drop, 128-135
- Nitrogen, film coefficients, 64  
pressure drop, 128-135
- Nitrous oxide, film coefficients, 64  
pressure drop, 128-135
- Nomenclature, for film coefficient equations, 56  
for pressure drop equations, 113
- Nozzles, 136, 137
- O
- Oblong coil (photograph), 155
- Octane (n), film coefficients, 64  
pressure drop, 122-134
- Oil heater, illustrative problems, 71, 121
- Open-type sections, 152
- Optimum thickness of insulation, 202-205
- Orifice-type baffles, 140
- Outlet from tube, pressure rise at, 113, 116, 117, 127, 132
- Over-all coefficient, chart for calculating, 44  
definition, 41
- Over-all coefficient, effect of scale on, 45  
relation between film coefficients and, 42-45
- Oxygen, film coefficients, 64  
pressure drop, 128-135
- P
- Packed floating-head, 142
- Parallel flow, definition, 47
- Passes, 138, 141
- Pentane, film coefficients, 64  
pressure drop, 122-134
- Perfect gas law, 58, 60, 116
- Petit, 34
- Petroleum oils, film coefficients, 61, 62, 105  
pressure drop, 114, 124
- Pipe, dimensions of (table), 216  
weight of wrought-iron, 216  
(*See also* Tubes)
- Pipe coils, 154, 155  
illustrative problem, 74-76
- Pipe covering, for cold lines, 176-182  
heat loss through, 200  
for hot lines, 163-172  
most economical thickness of, 203-205  
to prevent freezing, 205  
to prevent sweating, 206-208
- Plates, film coefficients for (outline), 64
- Powdered insulation, 195
- Pressure drop, calculation of, 113-135  
charts and tables, 122-135  
illustrative problems, 118-121
- Propyl alcohol (iso), film coefficients, 64  
pressure drop, 122-134
- Q
- Quilted asbestos blankets, 168, 195
- R
- Radiant energy, 1, 23, 30
- Radiation, 1, 23-36





Surface coefficient (*see* Film coefficient)

Sweating, prevention of, 206-208

Swirling strips, 139

## T

Tables, of combined coefficients, 200

of emissivities, 213, 214

of emissivity factors, 26

of film coefficients, 64

of gas densities, 130

of liquid densities, 122

of nomenclature, 51, 56, 113

of pipe dimensions, 216

of thermal conductivities, 209-212

of tube dimensions, 215

Tank suction heater, 146, 147

Tanks, method of insulating, 184, 188, 192, 194

Tape, insulating, 164, 170

Temperature, film, definition, 65, 118

at interface, calculation of, 14

of retaining wall, calculation of, 65

Temperature correction factor, 65, 118

Temperature difference, effect of, on film coefficient, 39, 41

mean, 46-52

in condensers, 46, 47

in crossflow heat exchangers, 47  
chart for calculating, 51

logarithmic mean (chart), 48

in multipass heat exchangers, 47  
charts for calculating, 49, 50

in single-pass heat exchangers, 47

Temperature gradient, in composite bodies, steady conduction, 14

for convection, 37, 66

in flat bodies, unsteady conduction, 17-19

Thermal conductivity, average temperature for, 7

definition, 3

effect of pressure on, 4

effect of temperature on, 4

Thermal conductivity, of insulating materials (table), 210, 211

of miscellaneous materials (table), 212

of metals (table), 209

units, 4

Thickness of insulation, most economical, 202-205

to prevent freezing, 205

to prevent sweating, 206-208

Toluene, film coefficients, 64

pressure drop, 122-134

Towers, method of insulating, 184, 192, 194

Tube bundle, partially assembled (photograph), 139

Tube bundles, equivalent inside diameter (chart), 95

film coefficients for (outline), 64

pressure drop, 117, 134

Tube-installation methods, 159

Tube lengths, 158

Tube materials, 158

Tubes, bowed, 138, 144

cleaning of, 137, 160

film coefficients for (outline), 64

finned, 138, 145

geometric data (table), 215

hairpin, 138

kinds of, 138

pressure drop in, 114-116, 122-133

replacement of, 138

U, 138, 143

Tube sheet, definition, 136

Tube-side fluid, definition, 136

Tube sizes, 158, 215

Tube wall, temperature of, calculation of, 65

thickness of, 158

Tubing, dimensions of (table), 215

Turbulence, effect on film coefficient, 39

Turbulent flow, 39

## U

$U$ , over-all coefficient of heat transfer, definition, 41

- U tubes, 138, 143
- Unsteady conduction, calculation of, through flat bodies, 17-22
  - definition, 5
- Use of charts and tables, 63-66, 118

## V

- Valves and fittings, insulation of, 172-179 181
- Vane-type baffles, 140
- Vapors, condensation of (*see* Condensation)
- Velocity, effect of, on film coefficient, 39, 40
  - on heat transfer, 113
  - on pressure drop, 113
- Vermiculite, 163 196
- Vertical long tube evaporator, 148

- Vertical short tube evaporators, 146-148
- Vibrators, 161
- Viscous flow (*see* Streamline flow)

## W

- Wall temperature, calculation of 65
- Waste-heat recovery, 152
- Water, film coefficients, 64
  - pressure drop, 122 134
- Water heaters, 155-157
  - illustrative problem, 69-71
- Weatherproof cement, 197
- Wellner-Jelinek evaporator, 150
- Whole sections, insulation, 163
- Wool felt insulation, 171, 181
- Wrought iron pipe, dimensions of (table), 216